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DESIGN AND WORKSHOP

DETERIORATION and destruction during the war, of factories and power stations, machines and tools, railways and ships, and of all kinds of industrial equipment, have caused a world-wide shortage of consumer goods and a lowering of the standard of living in many lands. With the exception of the United States whose production facilities remained intact, recovery is hampered by numerous economic and political factors. The gigantic programme the United States have embarked upon in agreement with the 16 nations participating in the Marshall Plan, and the aid they are also giving to facilitate economic recovery in the Far East, can without doubt greatly help in restoring prosperity and abundance. Nevertheless we must not lose sight of the enormous task which faces us to-day. American aid, however welcome, can never be more than a momentary relief, a bridge which may lead us back to prosperity, and only so if we take the best advantage of the opportunity it offers. The whole nation must be made aware of economic facts, unpleasant as they may be, and the initiative and inventiveness of the British people must be fully exploited.

Hardly a day passes in which we are not reminded of the ever-growing need for greater production. Our export trade is meeting more and more competition and the sellers' market is coming to an end, sooner than many had expected. As we are finding a growing reluctance amongst prospective buyers of British goods to accept the prices quoted by industry, we are asked, not only to maintain the present volume of exports, but to increase it considerably. Whereas, during the Battle of Britain and the years which followed it, the outcry was for greater output, more ships, more planes, and more tanks—whatever the cost—the Battle for Economic Survival, to-day and in the years lying ahead of us, demands not only greater production but an all-out effort for more finished goods produced in fewer man-hours and at a reduced raw material expenditure. Whilst all must work in harmony towards the one goal of National Recovery and Economic Stability, the Economic Survey for 1948 gave facts which called more to the workshop executive than to any other type of industrial worker. Although the slackness and absenteeism of some workers still accounts, in no small measure, for our present difficulties, we should pay tribute to those willing to do a full day's work by giving them the tools they so urgently need, and by co-ordinating their efforts so as to obtain the best results possible as regards quantity, quality and economy.

The sole purpose of this journal is to keep its readers informed on foreign engineering progress and developments. It is logical that at the present time when production is the key-note, our editorial work should lay stress upon articles of instructive value to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers and all those concerned with the practical aspects of engineering. Therefore, for the benefit of those engineers who plan and control the efforts of the man-on-the-machine, future issues of this Journal will include, in a new section "DESIGN AND WORKSHOP," a larger number of articles dealing with production organization, machine shop equipment, and new processes; and many other matters of interest to Production and Works Engineers will be presented with a view to introducing new methods which can lead to increasing output, and a reduction of capital expenditure and overheads.

This will in no way detract from the value of "The Engineers' Digest" to those whose interests tend more towards the academic and theoretical aspects of engineering, and for whom articles will be published as hitherto. Yet we feel that research-minded engineers wish to be kept up-to-date on the developments of engineering practice, and that some theoretical information can widen the practical man's outlook and enable him to deal more efficiently with the many important problems he encounters in his daily routine. Distrust and suspicion still exist between practical and research engineers. Let this be dispelled, and let the two co-operate to the full, and many stumbling blocks will be removed from the production line. Thus, in addition to stimulating the practical man's resourcefulness, "DESIGN AND WORKSHOP" should be one of the bridges leading to a better understanding between the "low-brow" and the "high-brow" and to a mutual respect for each other's work.

U. S. A.

Permanent Magnets of Sintered Oxides

By R. J. STUDDERS. (From *Product Engineering*, Vol. 19, No. 2, February, 1948, pp. 120-122, 6 illustrations.)

ALTHOUGH permanent magnets of sintered oxides have been known for some time, they have become available for industrial use only recently. Powder metallurgy processes and equipment are utilized extensively. The raw materials are fine oxide powders, totally minus 200 mesh, and of the type and purity widely used in the ceramic industry.

A typical mix is prepared by accurately weighing 31·0 per cent of red ferric oxide, Fe_2O_3, 43·0 per cent of black magnetic ferrosoferric oxide, Fe_3O_4, and 26·0 per cent of cobaltous oxide, CoO. These powders are intimately wet mixed in a porcelain jar mill for 36 hours.

The wet slurry is next dried at 160 deg. C until most of the water has been removed. A cake of material results from the drying operation and this is reduced to a powder again by forcing the cake through a 40 mesh screen.

The powder may be pressed to shape at this stage, but in order to control the shrinkage occurring during the sintering operation, the loose screened powder is usually presintered at 600 deg. C for 12 hr. in air.

Because of the nature of the oxide materials the sintered density is primarily a function of the pressed density. A high pressed density would seem desirable but, as the oxide powders do not flow well under pressure, high compacting pressures only serve to laminate the pressed compacts, rendering them useless. On the other hand, too low a compacting pressure results in a low density material of poor magnetic quality. No pressing lubricant or binder is used in the pressing operation, but careful attention is paid to the pressure used, particle shape, and moisture content of the powders.

The pressed compacts are fragile so that care must be used in handling them. After being placed in stainless steel boats, they are sintered at 1,000 deg. C for 3 hr. in an air atmosphere. Protection against severe thermal shock is necessary during the sintering operations. With proper regulation of the presintering operation, the linear shrinkage during sintering can be reduced to 1·5 to 2 per cent.

Similar to the more widely known Alnico 5, the permanent magnet material known as Vectolite is cooled through a region slightly below its Curie temperature in a magnetic field. A field strength of at least 1,000 oersteds is used. This treatment imparts a preferred axis of magnetization to the material, which is parallel to the axis of the magnetic field. When magnetized along the preferred axis, the material exhibits much higher magnetic properties than found along any other axis in the material. To illustrate this point: Curve 1 of Fig. 1 shows a typical demagnetization and available energy curve, taken with the specimen magnetized along its preferred axis. Curve 3 shows the demagnetization and available energy of the same specimen, but taken along an axis 90 deg. to the preferred axis. Curve 2 is a demagnetization and available energy curve for a specimen of sintered oxide magnet not cooled through the region below its Curie temperature in a magnetic field. From curves 1 and 2, it may be seen that the cooling in a magnetic field from elevated temperatures improves the residual induction, B_r, by some 45 per cent, the coercive force by 30 per cent and most important the maximum available energy by 150 per cent.

PHYSICAL AND ELECTRICAL PROPERTIES.

Differing from ordinary magnet materials, magnets of sintered oxides are semi-conductors and possess electrical resistivity, at room temperature, considerably higher than recorded for other magnet materials. Because of this property, Vectolite has found many applications in high frequency work. In permanent magnet applications where the practice has been to insulate the magnet material from its component electrical circuit, the sintered oxide magnet can be used without insulation. Fig. 2 is a curve prepared from data with a 1,000 v megger on a specimen of sintered Vectolite, previously dried at 200 deg. C. At room temperature this material appears to be practically an electric insulator.

Since Vectolite has a specific gravity of only 3·15, the demagnetization curve and the energy available compares more favourably with metallic magnet materials when made on a unit weight rather than on a unit volume basis. On the list of available permanent magnet materials, only Alnico 5, and modifications of it, show greater energy available than Vectolite per unit weight of material. This is significant since permanent magnets are usually priced on weight rather than on volume.

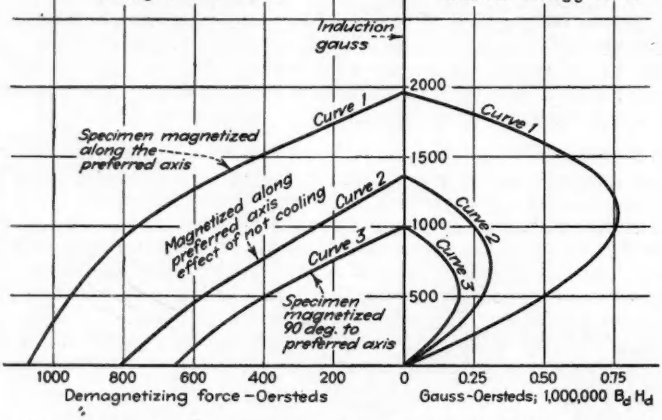


Fig. 1. **Demagnetization and external energy curves of specimens magnetized along preferred axis, perpendicular to preferred axis, and specimen not cooled in magnetic field.**

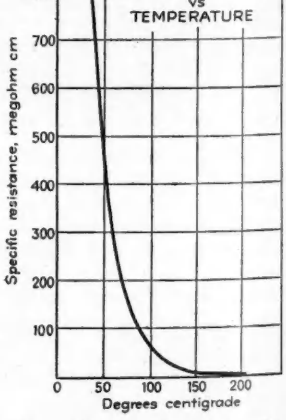


Fig. 2. **Variation of specific resistance of sintered oxide magnet with temperature.**

MAGNETIC PROPERTIES AND DESIGN DATA.

The design data, Table I, gives properties of the material magnetized in its preferred direction. Its anisotropic nature, while allowing the material to make more energy available, also imposes some limitations on its use by the designer. In manufacture, it is difficult to impart maximum properties along multi-directional flux paths by cooling in a magnetic field from elevated temperatures. Designs that must have abrupt directional changes in the flux paths through the magnet therefore should be avoided. Such magnets are expensive to manufacture. Care should also be taken to ensure that the final magnetization before use is in the preferred direction through the magnet. The loss of magnetic energy that would result from magnetization in a direction other than the preferred direction may be estimated by a comparison of curves 1 and 3 of Fig. 1.

TABLE I.—TYPICAL PROPERTIES OF SINTERED OXIDE MAGNETS ALONG THE PREFERRED DIRECTION OF MAGNETIZATION.

Magnetizing force for saturation, oersteds	3,500
B_s, Saturation induction, gausses	2,200
B_r, Residual induction, gausses (remanence)	1,900
H_c, Coercive force, oersteds	1,000
H_{ci}, Intrinsic coercive force, oersteds	1,150
$(B_dH_d)_{max}$, Maximum available energy, gauss-oersteds	750,000
at B_s gausses	1,000
Resistivity, Megohm cm at room temperature	See Fig.1
Coef. of linear expansion, 20 to 300 deg. C	$8{\cdot}5 \times 10^{-6}$
Specific gravity, grams per cu cm	3·15
Curie temperature, deg. C	540
Transverse modulus of rupture, psi	3,000
Max. operating temperatures, deg. C	200

A magnetizing force of at least 3,500 oersteds is necessary to saturate the sintered oxide material. Relative to the more common magnet materials a shorter magnet of larger cross-section will be needed to supply a given amount of flux. In common with all permanent magnet materials, the magnetic characteristics of Vectolite may be varied considerably by the method of manufacture.

In most commercial magnetic materials, the coercive force H_c and the intrinsic coercive force H_{ci}, are nearly the same. However, with material such as Vectolite, there is a considerable difference in these values of coercive force and the intrinsic coercive force of the magnet is a more accurate measure of the resistance to demagnetization. Data from an experimental determination of a demagnetization curve of a specimen of Vectolite, with some of the minor hysteresis loops, are shown in Fig. 3. For practical purposes the average slope of these minor loops is equal to the incremental permeability for the material. This quantity is important to the designer because the incremental permeability of a permanent magnet material is a measure of the degree of its stability against demagnetizing influences. The smaller the value of the incremental permeability the smaller the flux changes will be under demagnetizing influences. As may be seen from the graph, the incremental permeability is approximately one for Vectolite.

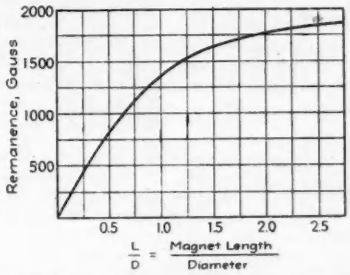


Fig. 4. **Variation of the flux density with the ratio of length to diameter of a straight bar magnet of sintered oxide.**

Flux density in the neutral section of a straight bar magnet plotted against the ratio of magnet length to its equivalent diameter is shown in Fig. 4. A curve of this type is used by many designers to predict the performance of a permanent magnet. This specimen possessed a residual induction, B_r, of 1,970 gausses, a coercive force, H_c, of 1,080 oersteds, and a maximum energy product. (B_dH_d) max. of 790,000 gauss oersteds at a flux density, B_d of 1,150 gausses.

HEAT, VIBRATION AND AGING.

To determine the maximum operating temperature of Vectolite and the effects of heat on it, a series of temperature tests were conducted as follows: Fully magnetized test bars after being heated to 100 deg. C and cooled to room temperature, showed less than 0·6 per cent loss in remanence. On successive cycles of heating to 100 deg. C no appreciable remanence change was detected. Heating to 200 deg. C and then cooling to room temperature, caused an initial loss in remanence of 2·4 per cent; successive cycles to 200 deg. C caused no further remanence change.

On initially heating a fully magnetized specimen to 300 deg. C a 28 per cent loss in remanence was detected on cooling to room temperature. This large loss is attributed to the destruction of the preferential alignment given the material by originally cooling it in a magnetic field. The Curie temperature for the sintered oxide magnet has been found to be about 540 deg. C, varying with impurity content.

From these tests the conclusion is that Vectolite may be safely operated to temperatures of 200 deg. C. Reasonable stability may be expected even if the magnet is heated a few times slightly above this maximum operating temperature. Cements that set below 200 deg. C. or resin binders that can be cured below 200 deg. C. may be safely used with the sintered magnet without destroying its magnetic properties. No appreciable change in its demagnetization curve after long periods of time, and little remanence change upon repeated heatings to 200 deg. C. indicate that the material is structurally stable in this temperature range.

To test the effects of vibration on Vectolite, fully magnetized test bars were vibrated at frequencies from 10 to 50 cycles per sec. with total displacements from 0·005 to 0·100 in. for as long as 90 min. Within the error of measurement no change in remanence of the test bars could be detected.

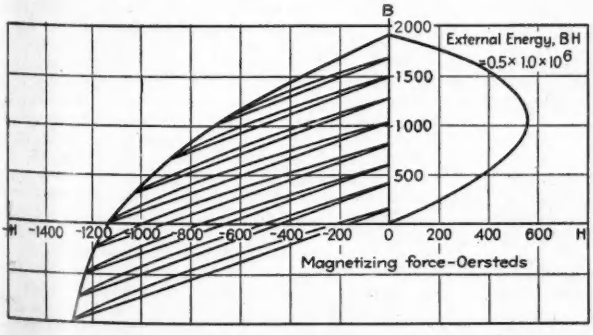


Fig. 3. **Minor hysteresis loops of a specimen of Vectolite.**

GERMANY

"Frozen Stress" Photo-Elasticity using Phenol Resins Wrapped in Protecting Foils

By E. MÖNCH. (From *Kunststoffe*, Vol. 37, No. 9, September, 1947, pp. 181-189, 16 illustrations.)

IN the frozen stress technique of photo-elasticity, a model of transparent phenol resin is loaded at an elevated temperature and allowed to cool slowly under load. It then sets and maintains the imposed strain with its photoelastic effect even after the load is removed. Finally the model is cut as required and the state of stress can be investigated optically at any desired point.

The disadvantages of this technique were, up to now, a relatively small stress-optical sensitivity and the so-called rind effect by which, due to a loss of moisture under heat, the outer skin tends to contract and sets up internal stresses which disturb the stress pattern. This rind effect can be prevented, however, by enveloping the resin model with a protecting layer of aluminium foil. Measuring accuracy is thus improved, the resin model can be cured before test, and the stress-optical sensitivity is increased threefold.

FUNDAMENTAL CONSIDERATIONS.

Phenolic resins can be assumed to consist of an elastic space lattice, of macro-molecules, filled by a plastic portion of material of lower molecular weight. The elastic network is independent of temperature but the plastic matrix, although fairly solid at room temperature, is largely fluid at about 80 deg. C.[1, 2, 3]. If the model is loaded at 80 deg. C. or more, the elastic lattice only is stressed, and if the material is cooled under load, before the load is removed, the "frozen" plastic matrix prevents an elastic recovery to the unstressed state. This behaviour corresponds to the B-stage during the condensation process of phenolic resins, and as the state of curing advances the elastic portion increases at the expense of the plastic. If the "B-material" is loaded at room temperature, the plastic matrix slowly gives way and pronounced creep ensues. For plane photo-elasticity, the model should therefore be as highly cured as possible. At elevated temperatures, the elastic portion only is strained, and the material is completely elastic without creep if the elastic lattice is strong enough to support the external forces. The modulus of elasticity at elevated temperatures is very much reduced, because only the elastic part of the material is active, and the optical effect is also reduced correspondingly[2], so that the stress-optical sensitivity is considerably smaller for the frozen-stress technique than for plane photo-elasticity.

Due to the rind effect, internal stresses are set up which distort the stress pattern near the model surface and reduce measuring accuracy. However, if the model is coated with oil and enveloped in aluminium or tin foil[4], it can be heat treated or preserved for several days in its protecting skin without the appearance of any rind stresses. In the frozen stress technique, the model can thus be cut out, machined, enveloped in foil, and heat treated in order to remove machining stresses and bring the material to a suitable stage of curing. The temperature at the time of loading and the cooling rate can be chosen independently of any considerations of the rind effect. The optimum curing stage and the temperature when loading have been found by the tests referred to in the following.

TESTS.

"Dekorit," the type of phenolic cast resin produced by Dr. F. Raschig, Ludwigshafen a. Rh., was used in all tests because it offered a high stress-optical effect and mechanical properties which could be widely varied according to the curing stage attained. For optical measurement, the stressed and cooled model was cut by bandsaw or, in some cases, turned. The increase in temperature, due to cutting, of medium hard Dekorit (cured for 6 hours at 115 deg. C.) was found to be not more than 8 deg. C. so that it could be assumed that the finished model could, under normal conditions, reach a temperature of 30 deg. C. (86 deg. F.). The material was essentially resistant to the effects of temperature up to that limit. The machinability of Dekorit decreases sharply, and leads to the production of rough surfaces and tool wear, when it is cured longer than 8 hours at 115 deg. C. (240 deg. F.) and this stage of curing should, therefore, not be exceeded. Also, the protecting foil loses its efficiency when the saturation pressure of the volatile materials in the resin is less than atmospheric. This effect was found to limit test temperature to slightly below the boiling point of water, unless tests are carried out at increased pressure. Tests were therefore conducted at temperatures up to 95 deg. C. (203 deg. F.).

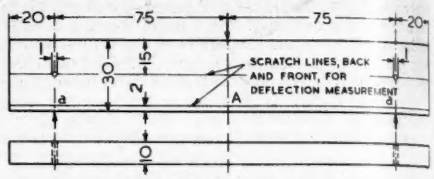


Fig. 1. Bending test-piece made from cast resin. (Dimensions in mm.)

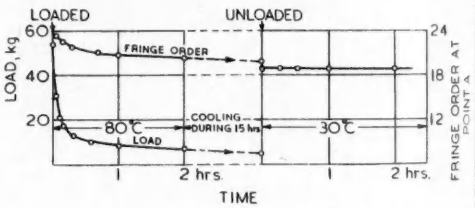


Fig. 2. Bending test by frozen stress technique. Loading temperature 80 deg. C., cure 7·09 hours at 115 deg. C. central deflection 4·2 mm.

The photo-elastic loading apparatus was arranged in a thermostatically controlled cell with special glass walls and, because of the high thermal expansion of the resin, the test specimen (Fig. 1) was arranged on sliding supports and loaded as indicated. The load applied and deflection at the centre were measured independently, and the deflection could be kept constant throughout the test. One hour of heating time was found to be sufficient to obtain uniform temperature throughout the material. The test-piece was cut from material which was machined all over to eliminate any preliminary rind effect. The test-piece was then wrapped in the foil and cured under increased pressure at 115 deg. C. (240 deg. F.)[4]. The optical effect was observed during the test and for this the protective foil was removed from the back and front of the test bar. After heating, the bar was loaded and the central deflection kept constant. After two hours under load at elevated temperature, the bar was cooled during 15 hours, re-heated to 30 deg. C., unloaded, and kept for 2 hours at 30 deg. C. The load and the fringe order at point A on the lower scratch line (Fig. 1) were measured throughout the test (Fig. 2). The fringes were obtained by circularly polarized monochromatic light and zero order could easily be detected by the symmetry of the arrangement (Fig. 3).

From load P and central deflection f, the elastic resistance, i.e. the modulus elasticity during test can be calculated by consideration of the influence of the bending moment and shear force (Poisson's ratio $m = 2$),

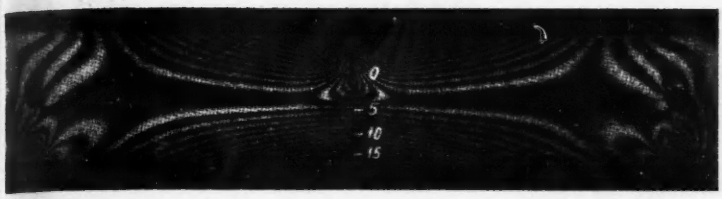
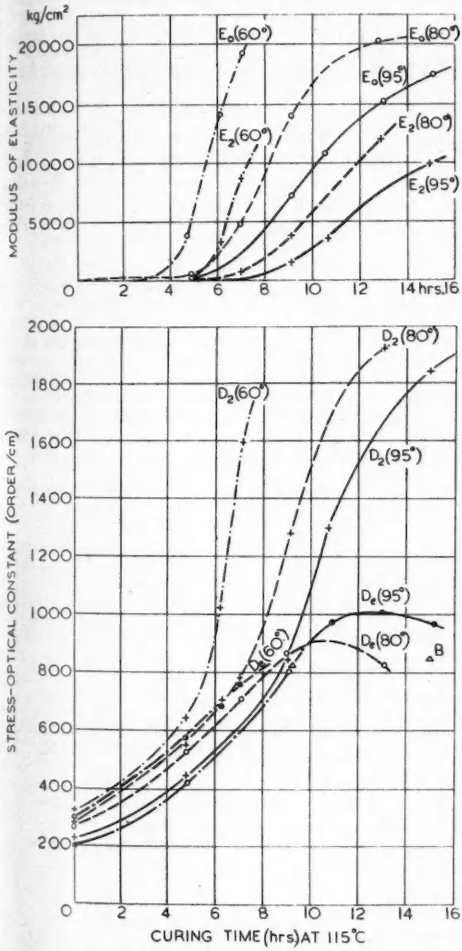


Fig. 3. **Fringe orders in bent bar, by normal photo-elastic method at room temperature.**
Model cured 4 hours at 130 deg. C., load 106 kg, deflection 1 mm. Circularly polarized light.

A calibration in pure bending also showed that one fringe order corresponds to 13·44 kg/cm², and, from Fig. 3, the stress at point A (Fig. 1) can be shown to be $\sigma_a = 2{\cdot}068\ P$ kg/cm², the strain being obtained from the ratio of stress to elasticity modulus. Finally, if the fringe order at point A is divided by the strain at A, the stress-optical constant D is :—

$$D = 17{\cdot}3\, n_A/e, \text{ where } E = 35{\cdot}75\, P/f \text{ kg/cm}^2;\ e = p_A/E$$



Figs. 4 and 5. **Bending tests by frozen stress technique, summary of results.**
E_0 Modulus of elasticity at the moment of loading ;
E_2 Modulus of elasticity 2 hours after loading ;
D_2 Stress-optical constant 2 hours after loading.;
D_e Maintained effective stress-optical constant, i.e. fringe order at end of test, 2 hours after unloading, referred to deformation held constant during test.

Fringe orders obtained by the frozen stress technique were compared point by point with those obtained by ordinary photo-elasticity procedure at room temperature (Fig. 3) and showed satisfactory proportionality.

TEST RESULTS.

Results from tests conducted as shown in Fig. 2 are represented in Figs. 4 and 5 as functions of the curing stage of the resin, and the temperature at the time of loading. Tests with uncured resin, as used up to now in the frozen stress technique, show, even at low temperatures, a small remaining stress-optical effect D_e. If the resin is cured, however, D_e increases appreciably, the more so, the higher the loading temperature. A limit is reached when the elastic resistance E_2, reaches a definite, but not negligible value just before cooling, because then the model recovers elastically when unloaded, and loses a great part of the optical effect, whilst the remaining internal stresses produce creep. Bearing in mind that a cure above 8 hours reduces the machinability of the model, the optimum conditions arrived at are :— cure for 7 hours at 115 deg. C. (240 deg. F.), loading at a temperature of 80 deg. C. (176 deg. F.), and a stress-optical constant of 705 fringe orders per cm, compared with 253 fringe orders per cm for uncured Dekorit, or 130 orders per cm for Bakelite BT61 893, as used by Hetényi at 115 deg. C. loading temperature[3]. The elastic proportionality, although not perfect, is satisfactory under these optimum conditions.

Longer test durations did not result in any improvements. Curing time and temperature are interdependent and a table* is given to show this. A check on the fracturing strength of Dekorit showed that the material was safe for all permissible stress-optical deformations.

PRACTICAL CONSIDERATIONS.

The spatial model is made from uncured Dekorit, machined all over (to remove at least 2 mm thickness), immediately coated with oil of medium viscosity, and covered with aluminium foil of less than 0·01 mm thickness. The foil should not tear and can advantageously be doubled if it is to be laid over a curved edge, e.g. of a drilled hole. (Fig. 6). For safety, a double layer should be arranged with oil between the layers. The model is then hardened, one face being placed in contact with a mirror glass plate for 7 hours

* Not reproduced here.—Editor, E.D.

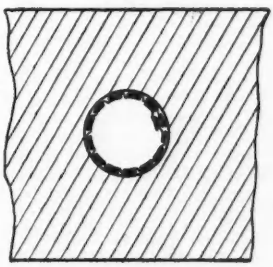


Fig. 6. **Protecting foil within the drilled hole (schematic).**

at 115 deg. C., or longer at lower temperatures. If the temperature is above 97 deg. C. but below 120 deg. C., a pressure of up to 2 atmospheres should be applied. A model showing considerable internal stresses may be preheated for 12 to 24 hours at 95 deg. C. The cured model is then placed in the test jig, heated at 80 deg. C. until temperature equilibrium is reached, loaded, and then locked in the loaded position. Folds in the cover, produced by deformation, should be smoothed to prevent air from entering beneath the cover. The temperature is kept constant for 2 hours, after which the apparatus is slowly cooled to room temperature and the model is unloaded. The foil is removed before cutting the model for measurement. High speeds and low feeds should be used for cutting, and if the surfaces are then coated with oil, they are generally transparent enough for measurement. Evaluation should be made by using convergent light[5].

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RUSSIA

The Influence of Heat Transfer between Cylinder Walls and Air upon the Operation of Piston-Type Compressors

By A. P. HERMAN. (From *Izvestia Akad. Nauk*, No. 12, 1947, pp. 1605-1614, 7 illustrations.)

IN the classical theory the working of a piston-type compressor of the dry type is represented by the diagram ABCD (Fig. 1) in which the distance V_1 represents the volume swept by the piston ; AB $= \lambda V_1$ is the volume of air drawn into the cylinder, where λ is the volumetric efficiency which is smaller than unity ; CD $= \mu V_1$ is the delivered volume and μ is the coefficient of delivery. It will be assumed that the condition of the air volume λV_1 is identical with that of the free air at the cylinder inlet. The capacity of the compressor Q_0 per minute is related to the free air volume by the relationship

$$Q_0 = i \lambda V_1 n \qquad .. \qquad .. \qquad (1)$$

where $i = 1$ in the case of a single acting machine, and $i = 2$ for a double acting compressor ; n is the number of revolutions per minute. The specific work per unit volume of the compressed gas in the case of polytropic compression is :

$$W_s = \frac{m}{m-1} p_s \left[\left(\frac{p_d}{p_s} \right)^{\frac{m-1}{m}} - 1 \right] \qquad .. \qquad (2)$$

As early as 1904 Kester* referred to the fact that the air admitted to the cylinder is heated by heat transferred from the cylinder walls, and he introduced the coefficient $\lambda_0 < \lambda$ so that the formula for the compressor output per minute can be written

$$Q_0 = i \lambda_0 V_1 n \qquad .. \qquad .. \qquad (1')$$

The ratio λ_0/λ is determined by the ratio of the specific volumes v_0 and v_1 of the free air and the air in the cylinder space respectively. For $p_s = p_0$ it reads :

$$\frac{\lambda_0}{\lambda} = \frac{v_0}{v_1} = \frac{T_0}{T_1} < 1 \qquad .. \qquad (3)$$

Very few test data is available for the determination of λ_0. A first attempt to determine λ_0/λ was made by Richter** in 1905 on a single stage compressor of a delivery pressure of $p_d = 8$ atmospheres for which he found $\lambda_0/\lambda = 0{\cdot}92$. Particular interest attaches to the determination of the λ_0/λ ratio of two-stage compressors for 7-8 atmos. delivery pressure. From test results obtained on a compressor without water-cooling of the cylinders it was found that for a delivery pressure of 7-8 atmos. it was $\lambda_0/\lambda = v_0/v_1 = 0{\cdot}936$, and $\lambda_0 = 0{\cdot}885$. In the case of water cooling the ratio λ_0/λ will of course be larger.

* *Zeitschr, V.D.I.*, 1904, pp. 109-118.

** *Zeitschr, V.D.I.*, 1905, pp. 1101, 1200, 1276.

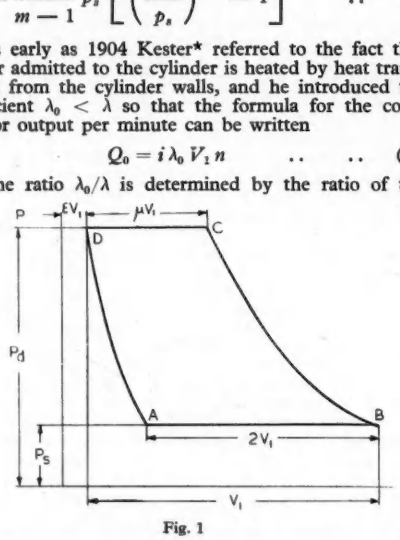


Fig. 1

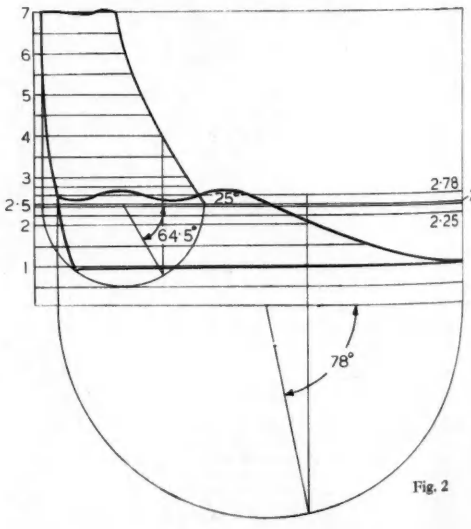


Fig. 2

The influence of the air velocity in the cylinder during the suction stroke will also have to be considered. In the case of a high piston speed, up to 3·5 m/sec, the heating effect will be intensified and for the low pressure cylinder it can be assumed that

$$\lambda_0/\lambda = v_0/v_1 = 0{\cdot}93 \quad .. \quad .. \quad (4)$$

For lower piston speeds of, say, 2 m/sec,

$$\lambda_0/\lambda = v_0/v_1 = 0{\cdot}94 \div 0{\cdot}95 \; .. \quad .. \quad (5)$$

It also follows that it must be $v_1 > \lambda_0$ .. (6)

With the use of the eqs. (4), (5), and (6) the capacity of the compressor can be established with sufficient accuracy for practical purposes, as will be exemplified by the following two cases. The first relates to an air compressor by Pokorny and Wittekind which had the cylinders arranged in tandem and was directly coupled to an electric motor. An indicator card obtained from this machine is shown in Fig. 2. The second example relates to a steam driven air compressor of the compound type built by Breitfeld and Danek, the indicator diagram of which is reproduced in Fig. 3. In the first case it is $\lambda_0 = 0{\cdot}88$ and $\lambda_0/\lambda = 0{\cdot}93$, and the weight of the air introduced into the cylinder per stroke is $G_1 = 0{\cdot}653$ kg and the delivery per minute is found as $G = 159{\cdot}2$ kg/min. In the second example, where $v_0 = 0{\cdot}88$ cu.m. per kg and $\lambda_0/\lambda = 0{\cdot}946$, it is found that $G_1 = 0{\cdot}564$ kg and $G = 76{\cdot}7$ kg/min.

Kester not only established the fact that the air receives heat from the cylinder walls during the suction stroke, but he also stated that in the case of insufficient cooling the compression process is first accompanied by heating until a certain point is reached beyond which compression is accompanied by cooling. This theory was accepted as correct by Prof. Ostertag† in 1910. By re-plotting the indicator diagram in the form of an entropy diagram, he obtained a compression line $A_1 A_2$ of the shape shown in Fig. 4 ; here it will be seen that up to the pressure p_x and the temperature T_x, the compression process is accompanied by heating. The author of the present report has found that the compression process is not, however, subdivided into two phases in the manner suggested by the classical theory, but that the subdivision into two phases is represented by a far more complex curve than that shown in Fig. 4. This is proved by the conversion of the indicator diagrams Figs. 2 and 3 into the corresponding entropy diagrams Figs. 5 and 6 which are obviously more complex than the diagram Fig. 4 given by Ostertag.

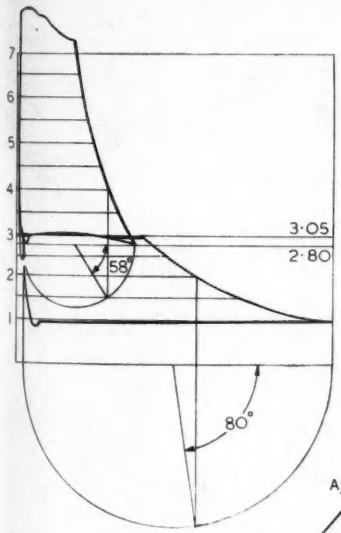


Fig. 3

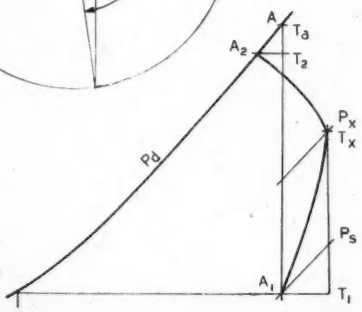


Fig. 4

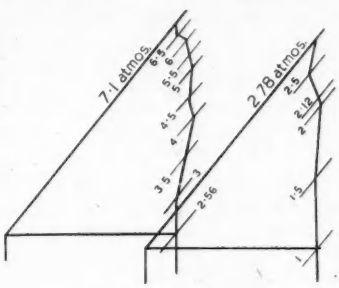


Fig. 5

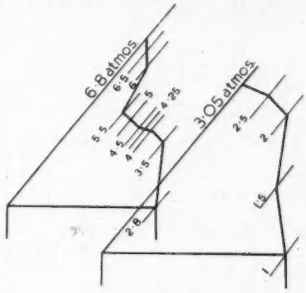


Fig. 6

The compression curves obtained do not as such appear to exhibit two phases, but the crank angularity relationships of these curves exhibit all the typical characteristics in this respect ; and, in fact, the two distinct zones showing positive and negative flow of heat q_2 and q_3 respectively during the compression period correspond with the first and the second phase respectively. These crank angles can therefore be termed the "angles of the first phase." For the Pokorny and Wittekind compressor the first phase is determined by an angle of 78 degrees, the corresponding heat flows being $q_2 = +\ 2{\cdot}0356$ and $q_3 = -\ 0{\cdot}874$ cal per kg. For the high pressure cylinder it is $q_2 = +\ 3{\cdot}3115$ and $q_3 = -\ 1{\cdot}8399$ cal per kg for an angle of $\alpha = 64{\cdot}5$ degrees. For the low pressure cylinder of the Breitfeld and Danek compressor it is $q_2 = +\ 0{\cdot}389$ and $q_3 = -\ 7{\cdot}317$ cal per kg, for a crank angle of 80 degrees. For the high pressure cylinder of this machine it is $q_2 = 0$ and $q_3 = -\ 1{\cdot}514$ cal per kg, the crank angle being 58 degrees.

Ostertag attributed the positive flow of heat during the first phase to the heat developed by piston friction. It is not difficult, however, to prove the erroneousness of this assumption, since the quantity of heat q_2 can surpass the heat q_r'' developed by piston friction, reckoned in terms of unit weights of air and based upon a full revolution of the crankshaft. Based upon a crank angle α the excess will be correspondingly greater. This goes to prove that the heating of the air during the first phase is much more complicated than suggested by Ostertag's theory; it is in fact caused by a complex heat transfer process taking place between the air and the cylinder walls. This heat transfer process has been analyzed by the author on the basis of Kirsch's theory of the flow of heat in the walls of a steam engine cylinder, and he has succeeded in establishing a general equation for the heat balance in the cylinder. He has also succeeded in establishing an equation for the amount of heat q_2 gained by the air during the first phase of the compression process.

† P. Ostertag: *Die Entropietafel für Luft*, 1910.

U. S. A.

Torque Converter Drive

(From *Machine Design*, Vol. 20, No. 3, March, 1948, pp. 122-124, 180, 7 illustrations.)

MATING of the internal combustion engine and the hydraulic torque converter to provide a drive of exceptional smoothness and flexibility has been so successful in the industrial field that ultimate adaptation to passenger cars has long seemed a foregone conclusion. First of the American passenger car makers to develop such a drive is Buick, which is now offering a torque converter drive called the Dynaflow, Fig. 1, as optional equipment on the 1948 Series 70 Roadmaster cars.

Characteristic of the torque converter is its inherent ability to provide torque multiplication as the ratio of output to input speeds decreases, Fig. 2. In a conventional transmission, or even in an automatic transmission this is done in three or four finite steps which must be manually or automatically selected. The torque converter, on the other hand, provides an infinite series of ratios which are self-adjusting at all times according to the relative engine and wheel torques. The Dynaflow provides a torque multiplication from one up to at least 2·24, which is adequate for most driving conditions.

Supplementing the torque converter is a planetary gear transmission under the control of the driver, who can select any of several alternatives—driving range, emergency low, neutral, or reverse. In the driving range the planetary unit provides a one-to-one ratio; in emergency low and reverse there is a reduction of 1·82 to one. When the torque converter is operating at its maximum ratio of 2·24 to one the overall torque multiplication in low and reverse is therefore 2·24 × 1·82 or about 4·1.

A longitudinal section through the complete Dynaflow transmission is shown in Fig. 1. The torque converter at the forward end is completely self-adjusting and requires no controls. Hydraulic fluid is circulated

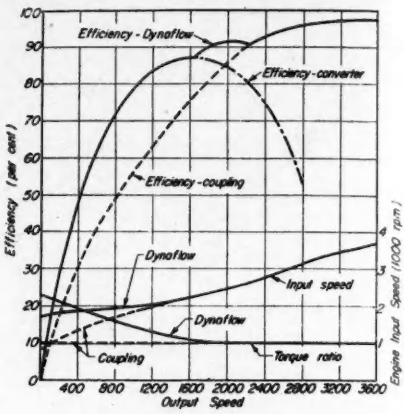


Fig. 2. Performance chart for Buick drive.

through it by either or both of two gear-type pumps. The planetary transmission incorporates two friction band brakes and one friction disk clutch, engagement or disengagement of which changes the speed ratio. Control brakes and clutches are actuated by hydraulic cylinder and piston mechanisms. The mechanical parking lock is operated by the same lever on the steering column which is used to select the desired forward, reverse, or neutral conditions.

The torque converter unit shown in Fig. 3 consists of five elements—a turbine, two stator elements, and two pump elements. The stator elements are necessary to absorb the torque reaction when operating as a converter, but are mounted on overrunning clutches which permit them to freewheel when the torque ratio drops to one. This occurs whenever the propeller shaft speed exceeds 85 per cent of the engine speed, and the converter then functions simply as a fluid

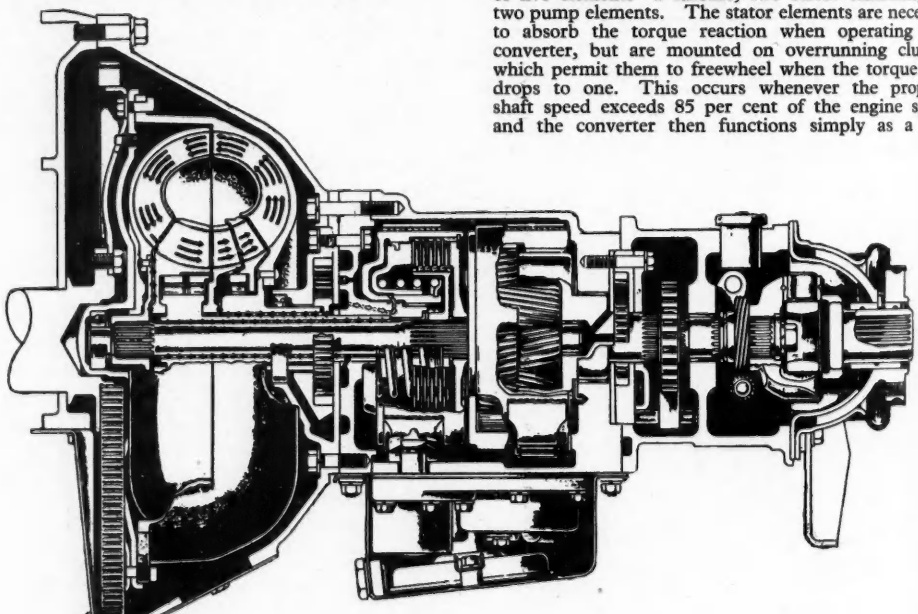


Fig. 1. Torque converter at forward end provides all changes in speed ratio required for normal driving. Planetary gear unit gives the driver control over emergency low, neutral and reverse conditions.

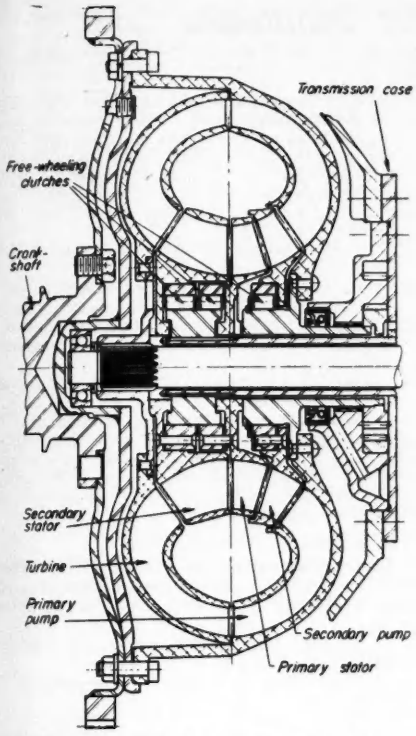


Fig. 3. **Cross-section through torque converter.**

coupling. Effect of this is illustrated in the performance chart, Fig. 2.

The stator vanes are curved so as to re-direct the fluid flow from the turbine back to the pump. At high torque ratios fluid pressure on the backs of the blades locks the stator elements to the transmission case through the action of the free-wheeling clutches, Fig. 3. As the speed difference becomes less, fluid pressure on the back of the secondary stator blades drops to zero, permitting the secondary stator to freewheel. As the turbine speed comes up to 85 per cent of the pump speed the fluid reaction forces on both stators drop to zero and both stators revolve at such a speed as to offer least resistance to fluid flow.

Somewhat similar action occurs in the pump. The secondary pump element is connected to the primary pump element through an overrunning clutch. At high torque ratios (large speed differences) the discharge velocity from the turbine through the stators is so high that it tends to push the secondary pump at a faster speed than the primary pump, which is directly connected to the engine. The freewheeling feature permits the secondary pump to overrun the primary and thus, as it were, get out of the way of the fluid and permit higher efficiency. As input and output speeds become more nearly equal and the velocity of fluid flow decreases, the pressure on the back of the secondary pump disappears and overrunning stops, the secondary pump becoming locked to the primary. The transitions from one stage of operation to another occur smoothly and gradually. The five torque converter elements are aluminium plaster-mould castings.

The planetary transmission unit is shown in cross-section in Fig. 4. There are two sun gears—the driving sun gear at the rear, which is splined to the input shaft, and the low-range reaction gear, which is connected to one element of a friction disk clutch. The other element of the clutch is splined to the input shaft, so that when the clutch is engaged the gear train becomes locked and the whole assembly rotates with the input shaft. The low-range reaction gear also is connected to a brake drum which can be locked to the housing when emergency low gear is required. The planet pinions are arranged in three pairs, each pair consisting of a low planet pinion, Fig. 4, engaging the driving sun gear, and a reverse planet pinion which engages (a) the low planet pinion ; (b) the low-range reaction gear, and (c) the reverse gear. The low planet pinions are approximately twice as long as the other gears to permit engagement with both the driving sun gear and the reverse planet pinions. In emergency low gear the forward brake band is applied, locking the low-range reaction gear to the housing. In reverse the rear brake band is applied, locking the reverse ring gear to the housing. In all cases the planet carrier is the driven member, except in neutral when the clutch and both brake bands are released and no driving effect results.

A hydraulic system is provided for the purpose of circulating fluid through the converter and oil cooler and for furnishing power to operate the clutch and brake bands of the planetary transmission. Pressure is maintained in the system by either or both of two gear-type pumps, the front one driven by the engine and the rear one by the propeller shaft. The front pump functions at starting, low speed, and in reverse; the rear pump is needed if for any reason the engine has to be started by pushing or towing. At speeds above 45 m.p.h. the rear pump takes over the whole job, the discharge from the front pump being simply recirculated. The complete transmission requires 11 quarts of a special oil, of which 8 quarts are contained in the converter while operating, the remainder being required for lubrication and for the hydraulic controls. The entire supply is continuously circulated through the oil cooler.

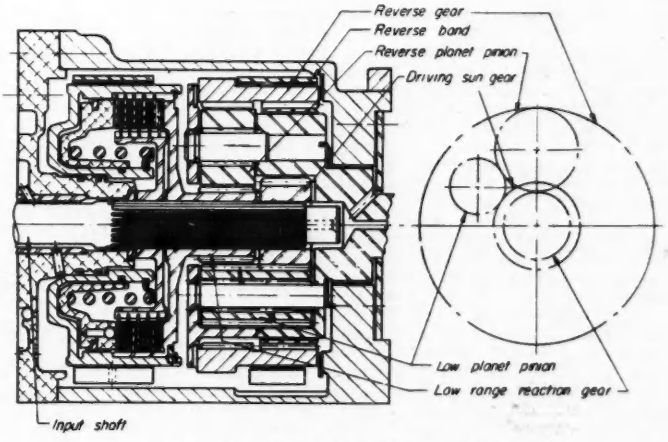


Fig. 4. **Cross-section through planetary unit.**

The Regulation of the Arc Suppressor Coil

By A. VAN GASTEL. (From *Brown Boveri Mitteilungen*, Vol. 34, No. 6/7, June/July, 1947, pp. 116-121, 12 illustrations.)

THE arc suppressor coil provides the best protection against the consequences of earth leakage, provided the coil current is adjusted in such a way that it is equal to the leakage current of the network (resonance tuning). In most networks, however, the earth leakage current is not constant, but varies according to the number, length and type of conductors under tension, so that the suppressor coil or coils must be correctly adjusted to obtain optimum results.

THE VARIOUS POSSIBILITIES OF ADJUSTMENT.

If each conductor which can be switched in or out of a network under normal operating conditions has its own arc suppressor coil, the adjustment of the coil current is completely independent of the geometry and length of the network, and is always correct. This applies only to networks consisting of a small number of lines of considerable length, i.e. to simple networks with comparatively high voltages. In the case of a network consisting of several lines, the total coil power required would have to be divided among a number of small low-power units and this would be an uneconomical solution. Consequently, a single coil is used in practice to protect a complete network or network section, and it can be adapted to various network circuits by an adjustment of the current. Three types of coils are used :—

1. Coils with tappings in the winding, the current adjustment being obtained with a tap-changing switch ;
2. Coils with tapped windings, and current adjustment in steps by means of a multi-contact switch ;
3. Coils with adjustable air gap and continuous adjustment of current.

The first solution is the least favourable of the three, because current adjustment with a tap-changing switch requires the coil to be separated from the network. A change of adjustment is then a complicated operation (opening of circuit-breaker, adjustment of switch, closing of circuit-breaker), which has to be performed at the place at which the coil is set up. Generally, this leads to the result that the service personnel soon omits these adjustments, particularly since a lack of adjustment is not at all noticeable under normal operating conditions.

A multiple-contact switch which can be operated under load simplifies the adjustment considerably, especially if it can be remotely operated by an auxiliary power unit from the control room ; but this equipment is expensive and requires considerable space for the step switch.

A favourable solution both from the technical and the economical point of view is obtainable with the arc suppressor coil with continuously adjustable current. This type of coil permits a particularly accurate adjustment the range of which can be made sufficiently large without much additional expense so that it can be used even when working in parallel with non-adjustable coils. As apart from the non-adjustable coils only those with continuous current adjustment have been recently ordered and produced, the following considerations will be restricted to coils of this type.

THE SUPERVISION OF COIL ADJUSTMENT.

If the exact magnitude of the leakage current of the network under supervision is not known, a remote indication in the control room of the current flowing through the suppressor coil will not suffice to show whether the coil current is correctly adjusted. This requires a measurement in order to determine whether the coil current corresponds to the capacitive leakage current, i.e. whether in the circuit constituted by the coil, the network, and the network capacity to earth, the inductivity of the coil and the capacity to earth compensate each other. Furthermore, if a small voltage exists in this circuit, then the resonance tuning of the coil can be obtained by :—

(a) a current measurement, determining the maximum current ;
(b) a voltage measurement, indicating when the voltage across the coil reaches its maximum value ;
(c) a measurement of the reactive volt-amperes of the circuit, determining when the reactive power of the circuit is zero ;
(d) measuring the power factor, and finding when $\cos\varphi$ equals unity.

The methods (a) and (b) have the advantage of only requiring simple measuring instruments for current and voltage determination. The network can be either symmetrical or asymmetrical (unequal partial capacitances of the three phases to earth); these two cases will therefore be treated separately.

A. ASYMMETRICAL NETWORK.

The asymmetrical network (see Fig. 1) is characterized by slightly different capacitances $C_{1,2,3}$ and leakage resistances $R_{1,2,3}$ and a voltage V_0 between

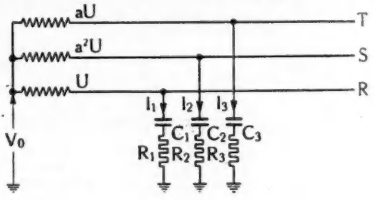


Fig. 1. Schematic diagram of a three-phase network with isolated neutral point.

$C_{1,2,3}$ = partial capacitances of the 3 phases relative to earth.
$R_{1,2,3}$ = leakage resistances of the 3 phases to earth.
U, aU, a^2U = symmetrical voltage system.
V_0 = asymmetrical voltage relative to earth.
$I_{1,2,3}$ = currents flowing across the partial capacities, with $I_1 + I_2 + I_3 = 0$.

the neutral point of the network and the earth. If the network is fed by three voltages of equal magnitude, U, a^2U and aU, with a relative phase of 120 degrees, then, using the abbreviations $W_k = R_k - j/\omega C_k$, the asymmetrical voltage will be given by

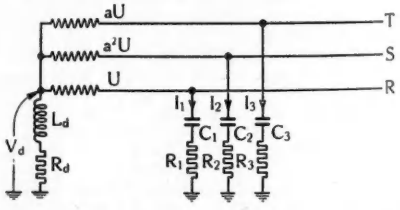


Fig. 2. Schematic diagram of a three-phase network with neutral point connected to the earth through an arc-suppressor coil.

L_d, R_d = inductance and leakage resistance of the suppressor coil.
$I_{1,2,3}$ = currents flowing across the capacitances $C_{1,2,3}$, with $I_1 + I_2 + I_3 = I_d$ (I_d = current flowing through the suppressor coil).
V_d = displacement current relative to earth.

$$V_0 = U \frac{\frac{1}{W_1} + \frac{a^2}{W_2} + \frac{a}{W_3}}{\frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3}} \quad .. \quad .. \quad (1)$$

The asymmetrical voltage is generally of the order of a few tenths of one per cent of the phase voltage U. However, as soon as a suppressor coil is connected and tuned to resonance (Fig. 2), the voltage between the network and earth will change considerably, and a displacement voltage V_d will then appear across the coil, which is determined by the expression (using the abbreviations $W_k = R_k - j/\omega C_k$, $Z_k = R_k + j\omega L_k$:

$$V_d = \frac{\frac{1}{W_1} + \frac{a^2}{W_2} + \frac{a}{W_3}}{\frac{1}{Z_d} + \frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3}} \quad .. \quad (2)$$

A comparison of eqs. (1) and (2) shows that the numerators are equal, so that the following relationship exists between V_0 and V_d:

$$V_d = V_0 \frac{\frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3}}{\frac{1}{Z_d} + \frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3}} \quad .. \quad (3)$$

As the earth leakage current is given by

$$I_e = U \left[\frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3} \right] \quad .. \quad .. \quad (4)$$

and the residual current (sum of the coil current and the earth leakage current) by

$$I_r = U \left[\frac{1}{Z_d} + \frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3} \right] \quad .. \quad (5)$$

the displacement voltage can also be written as follows :

$$V_d = V_0 I_e / I_r \quad .. \quad .. \quad (6)$$

When the coil current is adjusted to resonance, the residual current I_r has a minimum value and the displacement voltage reaches therefore a maximum value, which is not an absolute maximum but depends on the magnitude of the residual current. High leakage losses of the network, that is, high damping factors, are the cause of a large residual current and reduce the maximum of the displacement voltage. The curves of Fig. 3 show this maximum for leakage losses of 10 per cent and 5 per cent corresponding to 5 per cent and 10 per cent of the phase voltage. For a still smaller loss of 1 per cent, the maximum rises up to 50 per cent of the phase voltage. The same figure also shows how an improperly tuned coil, i.e. a coil adjusted to a current which is smaller or greater than the earth leakage current, causes a change of the displacement current. The curves are plotted against the percentage detuning ratio. A detuning of minus 20 per cent represents therefore an adjustment of the suppressor coil to 80 per cent of the earth leakage current.

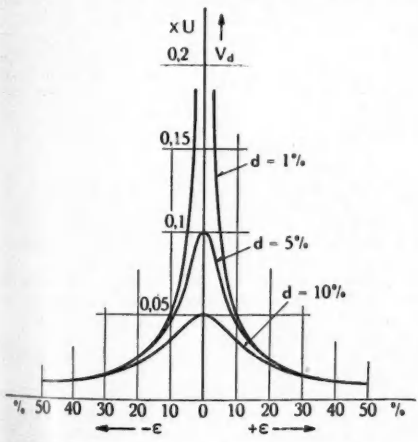


Fig. 3. **Displacement voltage V_d as a function of the percentage detuning ϵ of the suppressor coil for various values of the earth leakage of the network.**

The equipment required for the adjustment of the coil in an asymmetrical network according to the curves of Fig. 3 consists of two push-bottons for the remote control of the coil-regulating motor, a voltmeter, and a rotary switch for the switching-in of this voltmeter. The voltmeter has only a limited range, and to avoid overloading in case of an earth leakage, it is only switched on for the measurement ; the connexions are shown in Fig. 4. The voltmeter is switched on first to check the adjustment of the suppressor coil, and then the coil adjustment is varied with the push-buttons until the voltmeter deflexion reaches its maximum value.

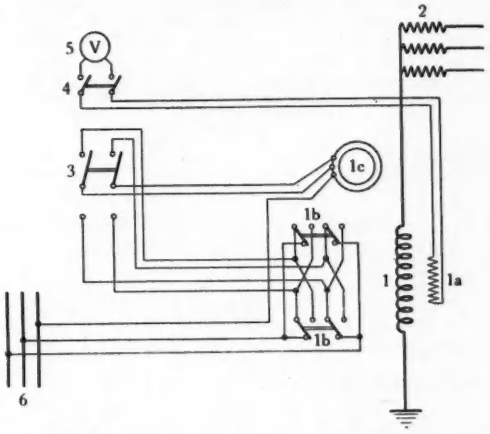


Fig. 4. **Circuit diagram for remote control of arc suppressor coil and adjustment using a resonance curve determined by means of a voltmeter.**

1 = suppressor coil ;
1a = additional voltage coil ;
1b = change-over switch ;
1c = regulating motor of suppressor coil ;
2 = transformer for connexion to suppressor coil ;
3 = adjustment switch;
4 = breaking switch ;
5 = voltmeter ;
6 = auxiliary source of energy.

Instead of a voltmeter for the measurement of the voltage V_d, an ammeter with a suitable transformer can be used to measure the asymmetric current in the circuit. The maximum current is then determined in the same manner.

For a measurement of the reactive power, a reference voltage is required. The resultant measurement current then is a mixture of two components, due to the asymmetric and the external voltage. An error would be introduced in the measurements if the resultant current were used in conjunction with the external voltage to measure the reactive power, and therefore, only the measurement current generated by the measuring voltage must be used, so that a filtering arrangement is necessary. The solution was achieved with the aid of

a frequency modulator. The considerable amount of equipment required in this case makes this method only economical in the case of fully-automatic operation. For the measurement of the power factor an external voltage and a filtering out of the measurement current are also necessary, and as these offer no particular advantages over the measurement of the reactive power, they will not be considered in this article.

B. SYMMETRICAL NETWORK.

In a symmetrical network the neutral point is at earth potential, as can be seen from eq. (1), where after putting $R_1 = R_2 = R_3 = R$ and $C_1 = C_2 = C_3 = C$, the numerator vanishes. In a network of this type the coil cannot therefore be adjusted by means of the resonance curve for voltage or current. This difficulty can be overcome by two methods :—

(a) by making the network asymmetrical ;
(b) by measuring with an external voltage.

With the first method a phase of the network is loaded by a resistor, an inductance or a capacitance relative to earth. If an ohmic resistance is used for artificial

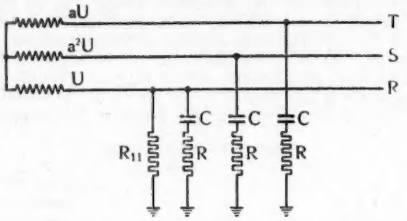


Fig. 5. **Earthing of a phase of the network through an ohmic resistance R_{11} in order to make the network asymmetrical to earth.**

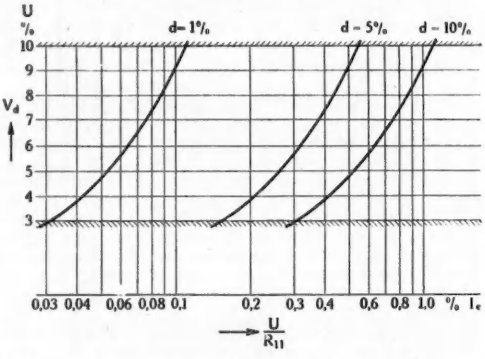


Fig. 6. **Displacement voltage V_d as a function of the resistance current U/R_{11} for the network which was made asymmetrical according to the circuit of Fig. 5.**

asymmetry (Fig. 5), there will again appear an asymmetrical voltage at the neutral point of the network, and its value is given by

$$V_0 = U \frac{1}{1 + (3R_{11}/W)} \quad .. \quad .. \quad (7)$$

where $W = R - (j/\omega C)$. After connecting the suppressor coil, the displacement voltage across the coil will be

$$V_d = U \frac{1}{1 + R_{11}\left[\frac{1}{Z_d} + \frac{3}{W}\right]} \quad .. \quad .. \quad (8)$$

where $Z_d = R_d + j\omega L_d$, the value of R_{11} being chosen in such a way that the displacement voltage will have a reasonable value. The displacement voltage should not exceed 10 per cent of the phase voltage, and not be less than 3 to 5 per cent of this voltage. As, however, the displacement voltage is damped considerably owing to the network leakage, the resistance values will differ considerably according to the degree of damping. Fig. 6 shows in the band between 3 per cent and 10 per cent of the displacement voltage, the resistance values for network dampings of 1, 5, and 10 per cent. The value of the resistor is given as a percentage of the earth leakage current, relative to the phase voltage. In determining the type of resistance it should also be borne in mind that in an earth connexion the resistor may also have to withstand the link voltage, so that a current which is $\sqrt{3}$ times greater may be passing through it. Its wattage should, therefore, be chosen three times greater. Similar conditions apply if a choke or a condenser is used instead of a resistor.

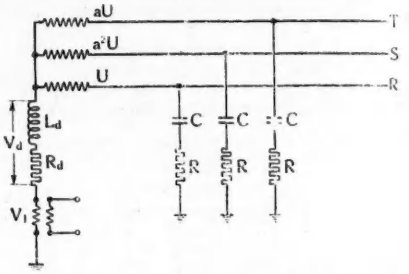


Fig. 7. **Neutral point fed with an external voltage V_1, at operating frequency, which is used instead of the asymmetrical voltage V_0.**

The results can be attained in a similar manner by feeding with an external voltage and using the circuit of Fig. 7, in which the external voltage V_1 is in series with the suppressor coil, between this and the earth. The displacement voltage V_d, which occurs when the external voltage V_2 is introduced, is determined by

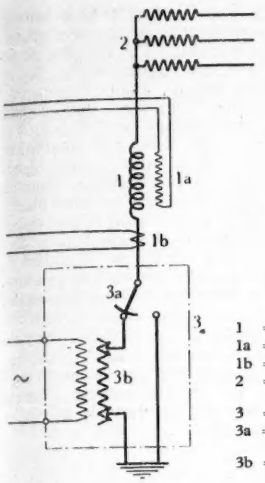
$$V_d = V_1 \frac{Z_d}{Z_d + (W/3)} \quad .. \quad .. \quad (9)$$

where $Z_d = R_d + j\omega L_d$ and $W = R - j/(\omega C)$. It can again be represented by an expression of the form $V_d = V_1 I_e/I_r$.

A comparison with eq. (6) for the asymmetrical network shows that the external voltage V_1 in this case assumes the rôle of the asymmetric voltage V_v. A small external voltage is sufficient ; for instance, with the coil adjusted to resonance, an external voltage equal to 1 per cent of the phase voltage of the network will cause a displacement voltage of 10 per cent when the network damping is 10 per cent, while with a network damping of 5 per cent an external voltage of 0·25 per cent of the external voltage will be sufficient to create a 5 per cent displacement. Furthermore, a variation in damping can easily be compensated by changing the external voltage, and for this only an adjustable source of current is needed, for instance, an induction type regulator or a variable voltage moving-coil type transformer for network regulation.

In networks which are only slightly asymmetrical the asymmetry can be increased by feeding the neutral point with an external voltage. The phase position of the external voltage relative to the asymmetrical voltage must then be chosen in such a way that the two voltages will sustain each other.

After introducing the external voltage the adjustment of the coil in the symmetrical network must be checked in the same way as for the asymmetrical network. In this case also the simplest method is to plot the resonance curve for the voltage or current and then to adjust to the resonance peak.



1 = suppressor coil;
1a = voltage winding;
1b = current transformer;
2 = main transformer for connexion to suppressor coil;
3 = control unit:
3a = change-over switch without breaker;
3b = auxiliary transformer.

Fig. 8. **Circuit diagram of the connexions to the auxiliary transformer supplying the external voltage V_1.**

Network	Additional aids required	Regulation based on	Remarks.
Asymmetrical	None	Resonance curve	Voltage or current curve nearly equivalent; simple method.
Symmetrical	Use of resistor, choke or capacitor	Resonance curve	Generally uneconomical on account of the high voltages for the additional equipment.
Symmetrical	External voltage	Resonance curve	Simple method.
Symmetrical	External voltage	Reactive power	Possible only when no asymmetrical current exists.

If the network really is symmetrically compensated then a reactive wattmeter can be used and adjusted to zero reactive power, with the external voltage used as reference voltage. The measurement will not be disturbed by an asymmetric current in this case, so that a frequency modulator will not be necessary.

The transformer supplying the external voltage used as reference voltage should be able to cope with the total coil current so that in the event of an earth leakage it will not be necessary to switch it off immediately. Nevertheless, a change-over switch working without breaking the circuit can be provided so as to have the possibility of adjusting at any moment to another tapping of the transformer. The circuit is shown in Fig. 8.

Summary of the Various Methods of Measurement.

The above table gives a review of the various methods of measurement :—

In overhead networks the asymmetry is nearly always sufficient to check and correct the adjustment of the suppressor coil by a simple plotting of the resonance curve.

In mixed networks (overhead lines and underground cables), and in underground networks, the coil adjustment can also be obtained by means of the resonance curve. If necessary the measurement voltage can be raised to an appropriate value by introducing a small external voltage between the coil and the earth.

BELGIUM

The Imperfect Film Lubrication of Sliding Journals

By L. Leloup. (From *Revue Universelle des Mines*, 9th Series, Vol. 3, No. 10, 1947, pp. 373-419, 47 illustrations.)

The efficiency of bearing lubrication is determined by oil film thickness. Theoretically, two different types of friction forces always occur simultaneously. They are (a) sliding friction between oil film laminae following the Newton-Navier hydrodynamic laws, and (b) tangential friction forces partly due to geometrical interpenetration of bearing surface irregularities and partly, or exclusively, to molecular attraction between surfaces. The thinner the oil film, the more important the second type of friction becomes. Knowledge of how the oil film is formed is therefore important for an appreciation of the actual friction conditions. Wedge forming surfaces (i.e. inclined thrust faces, and sliding journals with play) normally allow the formation of oil films of sufficient thickness for hydrodynamic lubrication, whereas parallel-slide bearings (i.e. slides, and pistons) usually produce much thinner oil films with a consequent preponderance of tangential forces. Nevertheless, even with wedge forming bearings, the oil film may fall below a critical thickness and the tangential forces become of increasing importance[1]. Only this last special case is considered in this article.*

If all other variables (i.e. pressure and viscosity) are kept constant, and only rubbing speed is diminished from an originally high value, the coefficient of friction first falls, in accordance with hydrodynamic laws, to a minimum value, but then rises again, contrary to hydrodynamic laws, and approaches a value of about 0·2 for zero speed. The ascending branch of the curve at low speeds characterizes imperfect film lubrication.

*The critical point of transition from hydrodynamic to imperfect film lubrication, the influence on imperfect lubrication of dimensional factors, condition and nature of bearing surfaces, and the properties of lubricants have been investigated by considerable research work in the machine design laboratory at Liège University. (See *Eng. Digest*, Nov., 1947.)

I. Influence of Dimensional Factors.

Z = absolute viscosity.
N = revolution per second.
V = peripheral rubbing speed.
p = specific pressure per unit area of projected bearing surface.
P = bearing load per unit of active bearing length.
R = bearing radius, d = bearing diameter.
r = shaft radius.
a = $R - r$ = radial play.
l = active bearing length.
f = coefficient of friction at the shaft, i.e. the "true" coefficient of friction.
f_c = coefficient of friction at the bearing, at small values different from f because, due to play, shaft and bearing axes do not coincide. However, f is always $< 2f_c$.

All values are expressed in units of kg, m, sec.

According to the hydrodynamic theory

$$f = f(X), \text{ where } X = \frac{ZN}{p}\left(\frac{r}{a}\right)^2.$$

Test experience confirms the theory[3], except for very small values of X for which f is larger than expected, when friction is then due not only to sliding between oil film layers but also to surface actions, and f increases even when X continues to fall. These two conflicting tendencies lead to critical values of Z, N, and p (r/a being constant) for which f is a minimum.

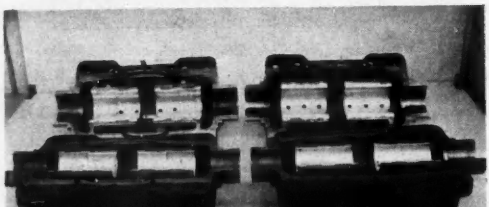


Fig. 1.

Tests were carried out on reamed white-metal bearings with shafts of 64,000 lb./sq. in. U.T.S. mild steel, turned to 1·58 inches diameter, and surface finished with emery cloth No. 1. Radial play was 0·0118 inches. The oil ring was fixed to the shaft and oil distributed into the upper bearing half. These bearings were run-in for about ten hours. A "long" bearing (l/d = 3·3, A = 8·3 sq. in.) and a "short" bearing (l/d = 1·65, A = 4·15 sq. in.), were used which were identical in all other dimensions except that the short bearing was slightly bored out at the ends to reduce its active bearing length,(Fig. 1). Normal machine oil, type "R" (for viscosity see Fig. 4), was used for lubrication. N and p were varied during tests, and Z changed according to ambient temperature, whilst the other factors were kept constant.

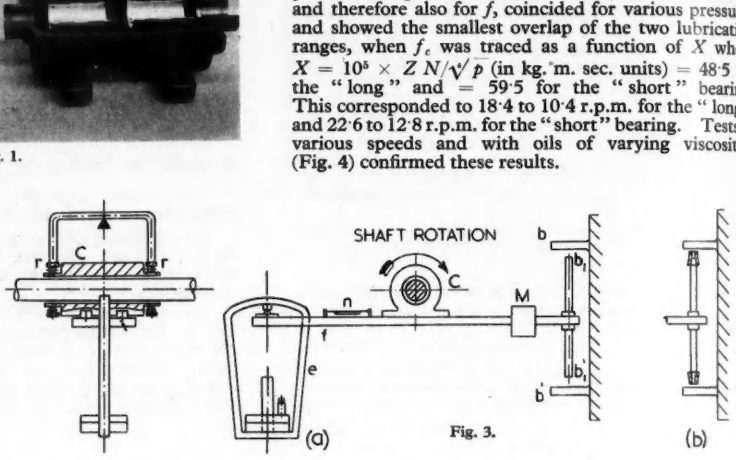


Fig. 3.

A. Critical Point.

Fig. 2 shows the arrangement, and Fig. 3 the test balance for the measurement of f_c. A weight placed on scale e provides the resisting couple, weight M is used for initial calibration, and level n determines the position of equilibrium. The balance is stable for hydrodynamic film lubrication since an increase of f_c tends to turn the bearing in the direction of shaft rotation and to diminishing the relative speed and, consequently, the friction f_c. Equilibrium is thus automatically restored. For imperfect film lubrication, the balance is metastable since reduction of relative speed leads to increased friction. The possible deviation of the bearing is therefore restricted (Fig. 3b) by rubber pads mounted on rod b_1, and a "dynamic" equilibrium is reached when both these pads deform an equal amount in alternating oscillations. The difference between stable and metastable equilibrium is very pronounced and permits fairly accurate determination of the critical point. Nevertheless, diagrams of f_c as a function of ZN/p, based on test results, showed a wide overlapping zone. Only when results were separated for each specific pressure, did a clear picture emerge. The critical values, for minimum f_c and therefore also for f, coincided for various pressures and showed the smallest overlap of the two lubrication ranges, when f_c was traced as a function of X where $X = 10^8 \times Z N/\sqrt{p}$ (in kg. m. sec. units) = 48·5 for the "long" and = 59·5 for the "short" bearing. This corresponded to 18·4 to 10·4 r.p.m. for the "long," and 22·6 to 12·8 r.p.m. for the "short" bearing. Tests at various speeds and with oils of varying viscosities (Fig. 4) confirmed these results.

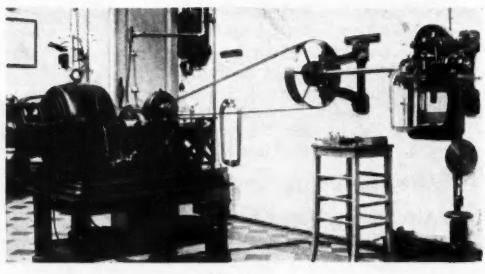


Fig. 2

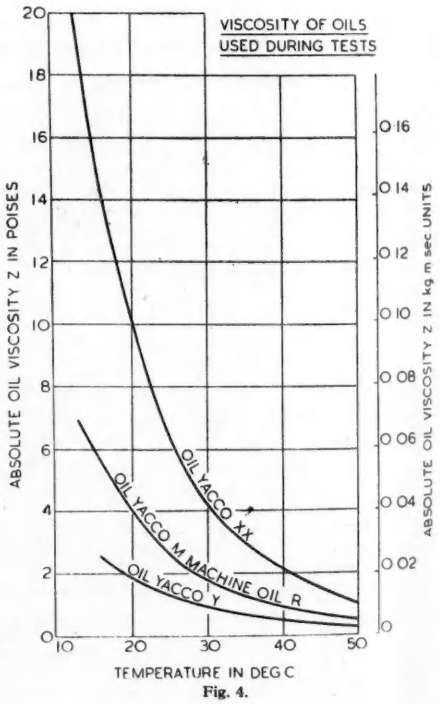


Fig. 4.

B. IMPERFECT LUBRICATION RANGE.

Under hydrodynamic conditions, the surface preparation is unimportant ; but for imperfect lubrication, it is very important. Each of the following series of tests was therefore restricted to 60 minutes duration, after which a new shaft was taken and run-in for some 10 hours. Pressures used were 42·5, 142, and 284 lb./sq. in. For f_c as a function of ZN/p, three curves were obtained, and these coalesced into one if f_c was traced as a function of $X = ZN/\sqrt[4]{p}$ (Fig. 5 is typical for the "long" bearing). For very small values of X, however, f_c rises more rapidly than by conformity to this formula, and the true law in such cases is $f_c = f(ZN)$, or even $= f(ZN\sqrt[4]{p})$ when near zero speed. For the main range of imperfect lubrication, as for the critical value, the general law, confirmed by tests with varying Z (Fig. 6) and N, is $f \simeq f_c = f(X) = f(ZN/\sqrt[4]{p})$ when R, a, l, the surface condition, and the type of oil are kept constant. f_c falls with increasing X until the critical point is reached. Figs. 5 and 6 show that $f_c = 0{\cdot}01$ for very small X. Under test conditions, with normal bearings, not very carefully prepared or run-in, using normal machine oil (i.e., 7·5 Engler at 50 deg. C.), $f_c = 0{\cdot}01$ at about 4·5 r.p.m. of the "long," and 7 r.p.m. of the "short" bearing, and the wide variations in pressure (under the fourth root in the equation) do not essentially influence the result. Below these speeds, f_c rises rapidly and approaches about 0·2 for starting conditions.

C. INFLUENCE OF R, a, and l.

Friction under hydrodynamic conditions, near the critical point, is independent of the play a/R. It can therefore be concluded that *play does not influence friction values in imperfect lubrication*, and tests confirm

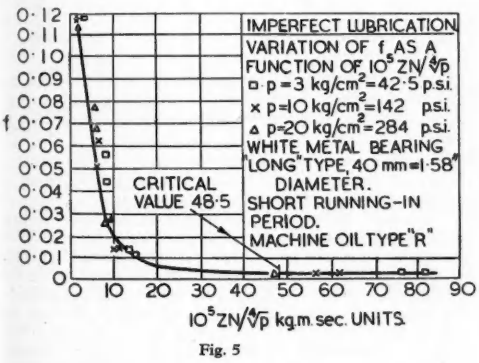

IMPERFECT LUBRICATION
VARIATION OF f AS A FUNCTION OF $10^5 ZN/\sqrt[4]{p}$
□ p = 3 kg/cm² = 42·5 p.s.i.
× p = 10 kg/cm² = 142 p.s.i.
△ p = 20 kg/cm² = 284 p.s.i.
WHITE METAL BEARING "LONG" TYPE, 40 mm = 1·58" DIAMETER.
SHORT RUNNING-IN PERIOD.
MACHINE OIL TYPE "R"
CRITICAL VALUE 48·5
f
$10^5 ZN/\sqrt[4]{p}$ kg.m.sec. UNITS.

Fig. 5

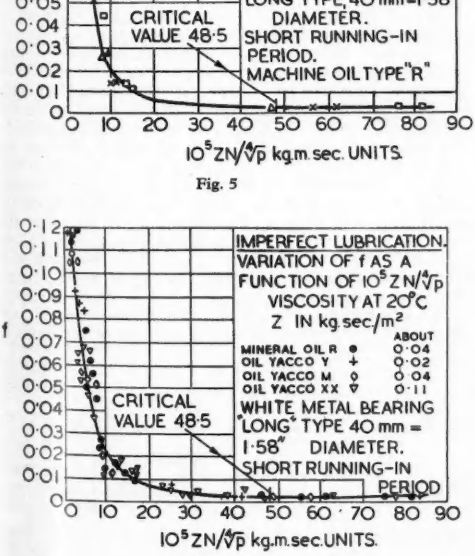

IMPERFECT LUBRICATION.
VARIATION OF f AS A FUNCTION OF $10^5 ZN/\sqrt[4]{p}$
VISCOSITY AT 20°C
Z IN kg.sec./m²
ABOUT
MINERAL OIL R ● 0·04
OIL YACCO Y + 0·02
OIL YACCO M ○ 0·04
OIL YACCO XX ▽ 0·11
WHITE METAL BEARING "LONG" TYPE 40 mm = 1·58" DIAMETER.
SHORT RUNNING-IN PERIOD
CRITICAL VALUE 48·5
f
ABSOLUTE OIL VISCOSITY Z IN kg m sec UNITS
$10^5 ZN/\sqrt[4]{p}$ kg.m.sec.UNITS.

Fig. 6

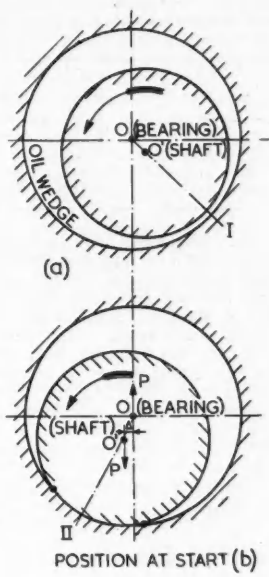


Fig. 7

this[4]. Specific pressure has no physical significance with imperfect lubrication because the true bearing area is smaller than the projected area, and conditions approach those of a thrust block, for which the hydrodynamic theory postulates $f = f(ZV/P)$. Similarly for imperfect lubrication

$$f = f(X_1) = f(10^5 ZV/\sqrt[4]{P}) = f(10^5 ZN/\sqrt[4]{p})\,\pi d^3.$$

X_1 is then valid for any diameter and its critical value is 16·7 for the "short" bearing. Similar values of 19·8 and 20·8 can be extracted from tests by other authors[4,5]. *The coefficient friction of imperfect lubrication falls with increasing diameter, N being constant.* Results for the two test bearing types coincide if

$$f = f(X_2) = f(10^5 ZV/\sqrt{Pd/l})$$

as a first tentative approach. The critical value for the test bearings is then $X_2 = 18{\cdot}5$. *With imperfect lubrication, a longer bearing shows less friction.*

II. INFLUENCE OF OIL AND SURFACE PROPERTIES.

The bearings were arranged in a closed chamber under controlled temperature. (See THE ENGINEERS' DIGEST, Nov., 1947.) Fig. 7a shows, schematically, relative shaft and bearing positions with hydrodynamic film lubrication, and Fig. 7b shows positions at starting-up and under imperfect lubrication. Traces of wear in bearings corroborate these relative positions. Apart from the friction couple, a couple $P \times \Delta$ (Fig. 7b) exists and, when f_c is very small, this is not negligible but adds to the friction couple for clockwise shaft rotation, and deducts from it for anti-clockwise rotation. The true value of f_c is then determined by the mean value of the couples measured in clockwise and anti-clockwise rotation. $P \times \Delta$ depends on P and should be constant if the bearing arrangement is not disturbed.

A. INFLUENCE OF OIL PROPERTIES.

Hydrodynamic theory assumes perfect adherence to the metal of the outermost oil film layers, and test results confirm this assumption by showing that the coefficient of friction is only influenced by oil viscosity

TABLE 1.

	Viscosity in poises at 20 deg. C.	at 50 deg. C.
Superanthralin (U.C.B.) ..	9·350	0·550
Pensylvanian oil	7·025	0·920
Mixed mineral oils	5·525	1·000
Glycerin	3·310	0·500
Vaseline oil	1·554	0·288
Anthralin (U.C.B.)	1·384	0·186
Syrup of sugar	0·962	0·220
Lard oil	0·924	0·263
Colza oil	0·830	0·260

(product $Z \times N$). In parallel-slide bearings, however, oils of equal viscosity give different lubricating properties according to their "oiliness."[8]. Lubricants of widely differing viscosity and oiliness were therefore tested in the sliding bearings shown in Fig. 1. (Table I). In each test series, any change in surface quality was excluded by reducing the test time, by a repetition of tests in inverse order, and by extrapolation of a mean test value. Under these conditions, the critical point is found to be independent of the type of lubricant used, and only viscosity matters. Other authors have given different results but their test conditions may not have been sufficiently controlled.[6,7].

In the lower range of imperfect lubrication, oiliness plays a definite role. Lubricants in order of decreasing oiliness, and correspondingly increasing coefficient of friction, are : animal oils (lard oil), vegetable oils (colza oil), mineral oils (machine oil, automobile-engine oil, anthracenic oils, and vaseline oils, in this order), glycerine, and syrup of sugar. The difference in friction values between lard and vaseline oils is only 20 to 33 per cent but glycerine gives twice, and syrup of sugar four times the friction of lard oil. The difference between oils is apparent below half the critical value of $ZN/\sqrt[4]{p}$, whilst syrup gives increased friction immediately below the critical point (Fig. 8). In this comparison of lubricants, viscosity changes have been excluded. Imperfect film friction is also always determined by viscosity or, more exactly, by the inseparable product ZN.

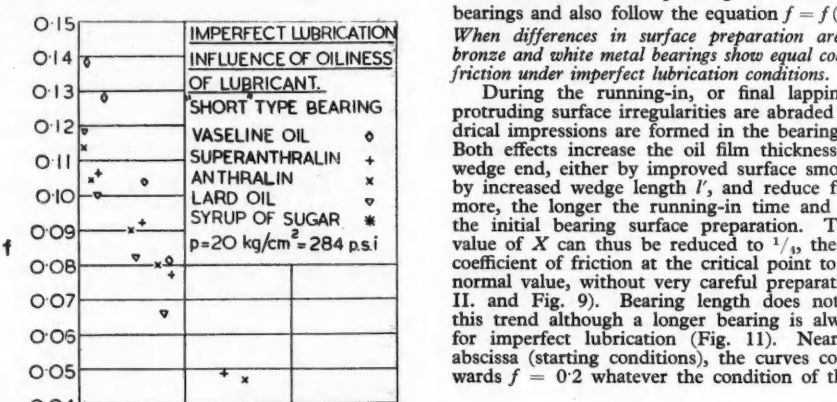


Fig. 8

B. INFLUENCE OF BEARING METAL AND SURFACE PREPARATION.

With hydrodynamic lubrication, the bearing surfaces are separated entirely by the oil film and cannot therefore influence the coefficient of friction. Tests on bronze bearings with imperfect lubrication (Fig. 9)

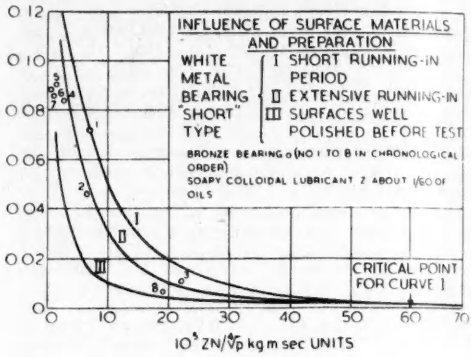

INFLUENCE OF SURFACE MATERIALS AND PREPARATION
WHITE METAL BEARING "SHORT" TYPE
I SHORT RUNNING-IN PERIOD
II EXTENSIVE RUNNING-IN
III SURFACES WELL POLISHED BEFORE TEST
BRONZE BEARING o (NO 1 TO 8 IN CHRONOLOGICAL ORDER)
SOAPY COLLOIDAL LUBRICANT Z ABOUT 1/60 OF OILS
CRITICAL POINT FOR CURVE 1
0·12
0·10
0·08
0·06
0·04
0·02
0
0 10 20 30 40 50 60 70
10⁵ ZN/∜p kg m sec UNITS

Fig. 9

TABLE II.

	Critical value $10^5\ ZN/\sqrt[4]{p}$	f_c per mil.
(1) Shaft turned, polished with emery cloth No. 1.		
(a) after 10 hours' running	48·5	1·800
(b) after several hundred hours' running ..	19·25	0·800
(2) Shaft lapped with cast iron rod.		
(a) as under 1b. ..	17	0·667
(b) after another 50 hours' running at 0·25 r.p.m.	12	0·600

show the influence of progressively increasing bearing perfection during test runs, but otherwise they arrange themselves on the corresponding curves for white metal bearings and also follow the equation $f = f(ZN/\sqrt[4]{p})$. *When differences in surface preparation are excluded, bronze and white metal bearings show equal coefficients of friction under imperfect lubrication conditions.*

During the running-in, or final lapping, period, protruding surface irregularities are abraded and cylindrical impressions are formed in the bearing (Fig. 10). Both effects increase the oil film thickness h_a at the wedge end, either by improved surface smoothness or by increased wedge length l', and reduce friction the more, the longer the running-in time and the better the initial bearing surface preparation. The critical value of X can thus be reduced to $^1/_3$, the minimum coefficient of friction at the critical point to $^1/_3$ of the normal value, without very careful preparation (Table II. and Fig. 9). Bearing length does not influence this trend although a longer bearing is always better for imperfect lubrication (Fig. 11). Near the zero abscissa (starting conditions), the curves converge towards $f = 0{\cdot}2$ whatever the condition of the surface.

However, if a bearing is run-in under a particular load, the condition of the surfaces is an optimum, and minimum friction results only for that same load. Fig. 11 shows extremes obtained during tests. Even if the low values of curve II are doubled in order to obtain

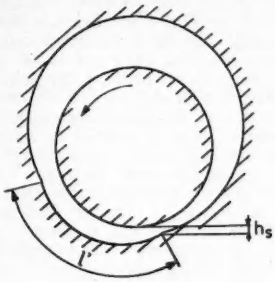


Fig. 10

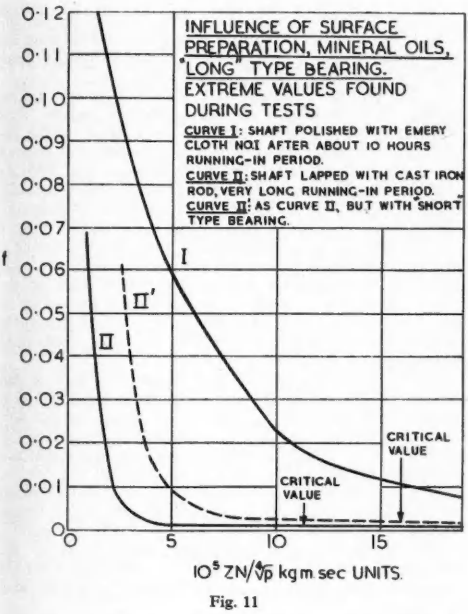


Fig. 11

the true coefficient of friction f, the *improvement due to better surface preparation is still paramount*. For the test bearing giving curve II, the coefficient of friction falls below 0·01 at 0·75 r.p.m..

III. CONCLUSIONS.

From the hydrodynamic theory of the thrust block and from the law that molecular attraction forces vary inversely with the square of oil film thickness, a general law for lubrication can be derived for wedge forming bearings

$$f = A\sqrt{ZV/P} + K\frac{1}{ZV\,l'},$$ which is valid in the

range $0 < ZV/P < 0{\cdot}03$. A and K are constant, l' is the wedge length (Fig. 10), f is a minimum when the two components are approximately equal, and the simplified formula $f = f(X_1) = f(10^5\, ZV/\sqrt[4]{P})$ represents a satisfactory approximation for the critical and subcritical range. For higher values of ZV/P, the hydrodynamic component prevails and $f = f(ZV/P)$. Above $ZV/P = 0{\cdot}03$, bearing play enters into the hydrodynamic considerations. For very small values of ZV, molecular attraction is preponderant and friction tends to be independent of the pressure P.

The better the surface preparation, the longer is arc l' and the lower is f for imperfect lubrication. Lubricants of high oiliness conform to the general law, and those of low oiliness, which therefore have reduced adherence to the metal, show higher coefficients of friction under imperfect lubrication conditions. The difference between various oils is not marked but other lubricants show considerably increased friction under imperfect lubrication. The influence of the ratio l/d cannot yet be explained. For starting conditions ($V = 0$), the general equation postulates $f = \infty$, actual conditions approach those of parallel-slide bearings without wedge effect, with f approximately 0·2.

The following *practical considerations* can be deduced :—

(1) In order to ensure perfect film lubrication of a sliding journal, the value

$$X = 10^5 \frac{ZV}{\sqrt[4]{pd/l}} \quad \text{should}$$

be larger than a critical value which, for a normal and fairly carefully prepared transmission bearing, without noticeable wear, is about 18·5. A safety factor of 3 ($X = 56$) is obtained in a sliding journal 1·58 in. in diameter, and 3·95 in. long, at 150 r.p.m. with a pressure of 21·3 lb./sq. in., and using machine oil DMC (R), $Z = 1{\cdot}249 \times 10^{-2}$ (kg. m. sec. units) at 35 deg. C. Since the coefficient of friction rises steeply only if X is below half the critical value, the actual safety factor against appreciable temperature rise or seizing is more than 6.

(2) If imperfect lubrication cannot be avoided at low speeds, the coefficient of friction should be reduced by careful preparation of the shaft and bearing surfaces. If, in extreme conditions, the shaft is lapped with a cast iron rod and the bearing is very well run in, a critical value $X = 5$ can be reached (as against 18·5) and the coefficient of friction reduced to $^1/_{30}$ of the value for normal bearing practice.

(3) For perfect lubrication, the bearing should be short ($l/d = 0{\cdot}8$ to 1), and for imperfect lubrication, a long bearing should be used.

(4) For starting conditions, a general coefficient of friction of about 0·2 can be assumed.

(5) Wear is noticeable with imperfect lubrication only and probably increases rapidly with increasing friction. Even if friction losses are unimportant, the coefficient of friction must be kept low if the bearing is to endure.

(6) White metal and bronze show equal coefficients of friction. White metal is normally preferred especially as the lapping of bronze bearings is more difficult.

(7) The oiliness of a lubricant is not very important even in imperfect lubrication. A good mineral oil is satisfactory, and organic oils offer no great improvement. Non-oily lubricants (glycerine) appreciably increase friction in imperfect lubrication.

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U. S. A.

X-Ray Thickness Gauge for Hot Strip Rolling Mills

By C. W. Clapp and R. V. Pohl. (From *Electrical Engineering*, Vol. 67, No. 5, May, 1948, pp. 441-444, 4 illustrations.)

The need for equipment to gauge continuously the thickness of the strip as it issues from the finishing stand has been obvious for some time. By ordinary gauging standards, the conditions under which the thickness measurement must be made are somewhat unusual. In a typical steel mill, the strip may be traveling at speeds of 2,000 feet per minute or more out of the finishing stand. As it leaves the rolls it may vibrate vertically through an amplitude of several inches and to prevent the possibility of jams or "cobbles," the clear opening through the gauge should be at least 18 inches high. The temperature of the strip is ordinarily between 1,500 and 1,750 degrees Fahrenheit. Cooling water sprays deluge the strip and any near by equipment.

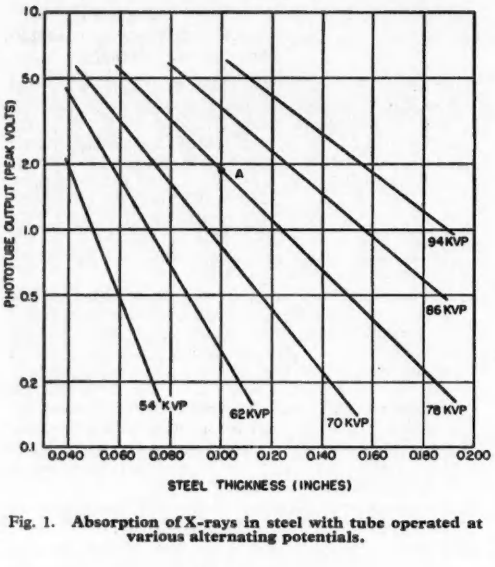


Fig. 1. **Absorption of X-rays in steel with tube operated at various alternating potentials.**

THEORY.

When a beam of X-rays strikes a sheet of some absorbing substance such as steel, a part of the beam is transmitted through the sheet unchanged while the remainder either is absorbed within the sheet or is reradiated as scattered X-rays.

A very satisfactory detector can be obtained by the use of a phosphorescent screen directly in front of a sensitive photoelectric tube. Such a detector was used to obtain the data for Fig. 1 which shows the intensity of the X-ray beam transmitted through different thicknesses of a standard low-carbon steel for several values of voltage V applied to the X-ray tube. In this instance, the tube was operated directly from a 60 cycle high-voltage transformer and V represents the peak value of the voltage across the tube during the half-cycle when the target is positive with respect to its cathode.

The curves of Fig. 1 are basic to the design of a thickness gauge for the material in question. They show for example that for a particular steel thickness and tube voltage represented by the point A, a 10 per cent decrease in tube voltage has the same effect on the detector as a 27 per cent increase in steel thickness. Effective means therefore must be found to stabilize or compensate for changes in tube voltage to secure a reasonable accuracy in gauge reading. The curves also show that for a given small change in steel thickness the corresponding change in detector reading is not constant but decreases in general as the thickness of steel is increased. This behaviour complicates the design of a gauge having a linear scale over a wide range of thicknesses.

DESCRIPTION.

As shown in Fig. 2, ports are provided in the enclosure for the X-ray tube to select two beams from the total radiation to be designated as the reference beam and the measuring beam, respectively.

The reference beam first passes through a long wedge of absorbing material (standard wedge) then falls on radiation detector 1. The output of the detector is a direct voltage whose magnitude and polarity are a measure of the difference between the intensity in the reference beam and a standard reference intensity established within the detector for comparison purposes. The output voltage of this detector is used to control an electronic voltage regulator in the primary circuit of the high-voltage transformer supplying the X-ray tube. By thus automatically controlling the voltage on the X-ray tube, the intensity in the reference beam is maintained at a level such that, after partial absorption in the standard wedge, it will have the value determined by the reference standard in detector 1.

The measuring beam first passes through a thin wedge of absorbing material (measuring wedge) then through the thickness of steel to be measured and falls on the radiation detector 2. Again the output of this detector is a direct voltage measuring the difference between the intensity of the beam striking the detector and a standard reference intensity established within the detector for comparison purposes. The output voltage of this detector, in turn, is used to control a motor geared to the measuring wedge. When the intensity in the measuring beam is different from the reference intensity, the motor shifts the wedge in a

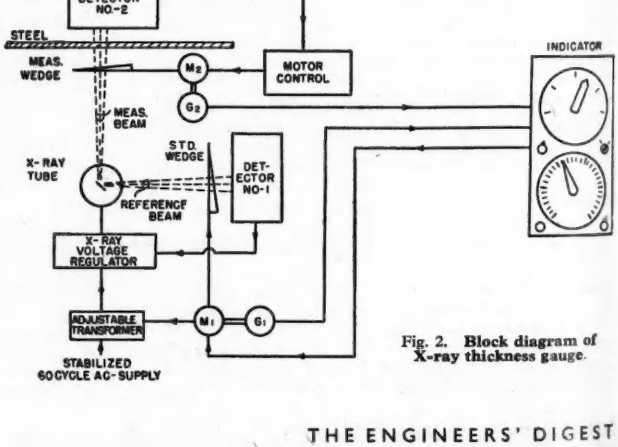


Fig. 2. **Block diagram of X-ray thickness gauge.**

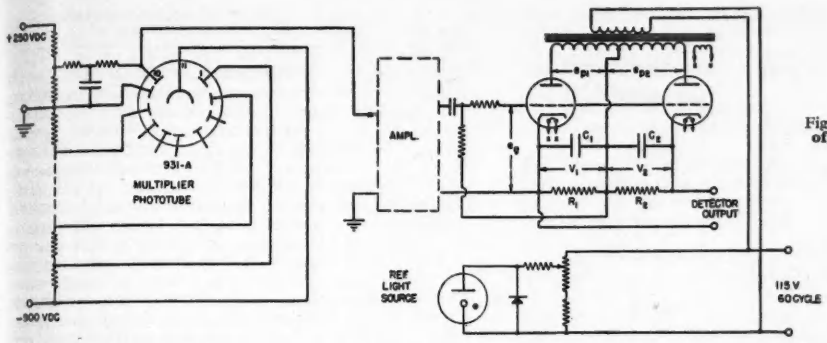


Fig. 3. **Circuit diagram of radiation detector.**

direction to increase or decrease the absorption in the beam as required to maintain the intensity at the value determined by the reference standard in detector 2.

In operation, the reference standards in the two detectors are adjusted so that the measuring wedge comes to rest when the total absorption in the path of the reference beam equals the total absorption in the path of the measuring beam. By making both wedges of steel, the thickness of the unknown steel sample can be obtained simply as the difference between the effective thickness of the two wedges. To provide a simple means of reading the gauge, the positions of the two wedges are transmitted through separate Selsyn systems to two scale pointers on a remote indicator panel. One of these pointers, connected to the standard wedge, indicates the nominal steel thickness to which the gauge is set. The second pointer, connected to the measuring wedge, indicates on a zero-centre scale the deviation in sample thickness from the nominal gauge size.

To set the gauge to a different value of nominal steel thickness, the standard wedge must be moved to present a different thickness to the reference beam. This is done by means of a small reversible motor controlled by push buttons on the remote indicator panel. The same motor also is linked through a cam mechanism to the arm of a variable-ratio autotransformer connected in the primary of the X-ray high-voltage transformer. As the standard wedge thickness is altered, the voltage on the X-ray tube thus is adjusted automatically to the approximate value needed to maintain the required intensity in the reference beam striking detector 1. The range over which the electronic voltage regulator must supply correction is thus reduced greatly.

RADIATION DETECTOR.

The circuit of the radiation detector is shown in Fig. 3. This employs a 931A multiplier-type photoelectric tube with a thin layer of silver-activated zinc sulphide phosphor cemented to the outside of the tube envelope opposite the cathode surface. When excited by X-rays, this phosphor emits light having most of its energy at a wave length of 4,350 angstroms. Since the 931A has its peak response at 4,200 angstroms, this makes a reasonably sensitive combination. Photoelectric tubes of this type vary widely in sensitivity but average tubes so coated have been found which, when placed in a beam of 50 milliroentgens per hour, gave a signal 30 decibels above tube noise. The sensitivity of the multiplier-type photoelectric tube varies rapidly with changes in electrode voltage, and is subject to fatigue effects. These disadvantages have been overcome by requiring only that the photoelectric tube compare the relative strengths of two light signals and indicate which of the two has the greater intensity. The first of these signals results from the excitation of the phosphor by the X-ray beam and consists of half-wave pulses occurring at line frequency. The second light signal is obtained from a glow discharge lamp of a type recently developed for light-modulator service. This lamp is connected to the 60 cycle line through a ballast resistor, and is shunted by a diode rectifier to ensure its excitation only on the alternate half-cycles of line voltage when no X-rays are being emitted. The light from this lamp, after being suitably attenuated through filters, is directed through the phosphor onto the cathode of the photoelectric tube.

To compare the two peak values, the signal from the photoelectric tube first is amplified through a linear amplifier, then applied to the grids of the phase-sensitive triode detectors. In this circuit, the plates of the two detector tubes are supplied with voltages of equal amplitude but opposite phase through a centre-tapped transformer connected to the 60 cycle line. The cathode load resistors R_1 and R_2 are made very large compared with the reciprocal of the mutual conductance for the triodes and the capacitors C_1, C_2 are selected so that the time constants $R_1 C_1$ and $R_2 C_2$ are long compared with the period of the signal wave. Under these conditions the capacitors C_1 and C_2 charge up to steady voltages V_1 and V_2 sufficient to maintain their respective tubes close to or beyond plate current cut-off at all times. Because the plate voltage on one of the triodes, say e_{p1}, has its maximum value when the signal from the X-ray beam is a maximum, and the plate voltage on the second tube (e_{p2}) reaches its maximum in phase with the signal from the reference light source, V_1 and V_2 are linear functions of the peak values of the X-ray signal and the reference light signal respectively. The output of the detector, consisting of the voltage $(V_2 - V_1)$, is therefore zero when the two signals have equal peak values and at other times has a value proportional to the difference between the two signals.

ELECTRONIC REGULATOR.

The circuit diagram for the electronic regulator used to control the accelerating voltage on the X-ray tube is shown in Fig. 4. This regulator operates by varying a resistance element connected in series with the primary of the high-voltage X-ray transformer. The resistance element consists of the plate resistance of four 807 tubes connected in parallel. The transformer T_2 serves to provide a proper impedance match between the plate circuits of the tubes and the power line circuit into which they operate. To utilize the 807 tubes efficiently, their screens are maintained at a steady direct potential. The grid bias on the tubes is controlled by the output of radiation detector 1, amplified through one stage of d.c. amplification. When connected as shown the tubes carry current only on one half-cycle of the line voltage, this of course being the half-cycle during which the X-ray beam is emitted. The resistor serves a dual purpose; it limits the voltage appearing across the tubes on the negative half-cycles of voltage and during the warm-up period of the tubes, and it increases the regulating range of the circuit.

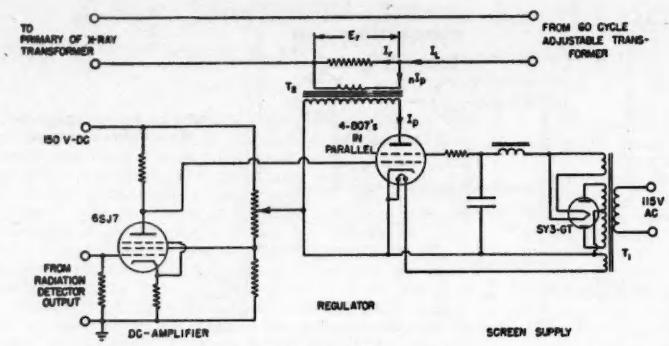


Fig. 4. **Circuit of electronic voltage regulator.**

OPERATING PERFORMANCE.

The X-ray tube and associated high-voltage and filament supply are housed in a single unit and are contained in a lead-lined oil bath, with forced oil circulation to the anode of the tube and separate water cooling of the oil. The tube used has a tungsten target and is rated at 100 kV peak. Also contained in this unit are the two steel wedges, with their associated motor drives and connected Selsyn transmitters, and the photoelectric tube and reference light source for detector 1.

The photoelectric tube and reference lamp for detector 2 are mounted in a separate water-cooled steel housing. At installation, this unit is mounted about 24 inches above the lower unit and is centred in the measuring beam which is directly vertically upward from the lower unit.

A cabinet houses the remainder of the electronic detection and control equipment as well as the power supplies for all circuits.

A remote indicator shows the nominal gauge size in thousandths of an inch and the deviation of the steel thickness in thousandths of an inch from the nominal size. Means are provided at the indicator station to correct the calibration of the gauge when necessary and to select any nominal gauge size within its range.

There are certain points that should be considered in using a gauge of this type which are related to the operating conditions. Among these are the effect of impurities or changes in composition of the steel, the additional absorption of the layer of cooling water retained on the steel strip, and the contraction of the steel in cooling down to room temperature.

Fortunately, most of the alloying elements commonly used in making steel have an X-ray absorption coefficient less than or only slightly greater than that of iron. Tungsten and molybdenum are the principal exceptions to this statement, but are not commonly used in sheet steels. Stainless steel of the common 18-8 composition has an absorption coefficient within one per cent of that of iron at room temperatures and this difference readily can be taken care of by a zero adjustment during calibration. For common grades of silicon steel the correction may amount to seven per cent. The gauge dial of a gauge used for measuring several alloys has several concentric scales. Puddles of cooling water an eighth of an inch thick frequently may be found on the strip at the measuring point. Such a layer can cause an error in the order of two per cent. Where this is objectionable, the water can be removed from the measuring point by a strong air blast.

While the steel is being gauged, its temperature is in the neighbourhood of 1,600 degrees Fahrenheit. As compared with the same steel at room temperature, the hot strip presents fewer iron atoms in a given cross-section of the X-ray beam and hence gives a lower reading of thickness. The correction for temperature depends only on the expansion coefficient of the steel, and for ordinary carbon steels amounts to 1·6 per cent per 1,000 deg. F. As the rolling temperature usually is known to within plus or minus 200 degrees, it is a simple matter to graduate the scales to correct for this error.

SAFETY CONSIDERATIONS

In a device of this type intended for use by non-technical personnel, the question of safety must be given full consideration. First, the unit containing the X-ray tube has a fully protective housing. Second, by using sensitive detection means, the X-ray intensity needed in the measuring beam has been kept as low as possible. Third, warning lights and barrier gates interlocked with the X-ray tube voltage supply have been provided at all points of access to the gauging area to prevent personnel from entering this area while the beam is on.

U.S.A.

Engineering Problems in Atomic Energy for Industrial Application

By J. A. HUTCHESON. (From *Westinghouse Engineer*, Vol. 8, No. 2, March, 1948, pp. 37-38.)

No one questions that it is technically possible to achieve the controlled release of atomic energy in a form that can be converted into heat or electricity. However, before this is actually an accomplished fact there is a great amount of work to be done. This work falls into two general classifications: namely, research and engineering development. At present there is under way in the United States a research program, co-ordinated and directed by the Atomic Energy Commission, which, as time goes on, will uncover fundamental knowledge of this new science that is essential before industrial applications of atomic energy are practical. Undoubtedly therefore, as the store of available knowledge increases, there will come a time when answers to the many questions relating to practicality of atomic energy in industry can be determined with reasonable certainty. What then are the engineering problems that must be solved before the first practical electric power generating station can be built?

The method by which a chain-reacting pile or nuclear-reactor operates has been widely discussed. It is known, for example, that when a neutron, whose kinetic energy lies within a certain range, strikes the nucleus of a uranium-235 atom in such a way as to be absorbed, fission of the U-235 atom results. Products of that fission are two elements each of substantially lower atomic weight, gamma-ray radiation, the discharge of one to three neutrons, and the release of extremely large amounts of energy. Further, some of the neutrons released eventually strike the nuclei of other U-235 atoms, whereupon the process is repeated, i.e., a chain reaction is established. However, the chances

of a neutron striking the nucleus of another atom of U-235 are subject to several factors. Obviously other kinds of atoms are present in a reactor. For example, there are those of the reactor coolant or heat-transfer medium, the structural material and surrounding shielding, those of the many control elements and devices essential to the operation of the reactor, as well as atoms of the many other elements that are the products of previous fission of U-235 atoms.

A few materials are partially "transparent" to neutrons. In some, such as carbon, absorption of a neutron is relatively unlikely. A neutron that enters such a material may collide with atoms of the material and be slowed up or lose part of its energy as a result, but in general it does not combine with these atoms and therefore is not absorbed. Again, the chances of absorption of a neutron depend in a complex way on the energy of the neutron. For example, it is much more likely that a neutron possessing an energy of a few tenths of an electron-volt will be absorbed by the nucleus of a U-235 atom with resultant fission than a neutron of a few hundred or more electron-volts energy.

These facts were employed in designing the reactors at Hanford, Washington. In these reactors the fission-produced neutrons released with high energy are slowed down by collisions with atoms of carbon in the moderator to the point that the average energy of the neutrons that re-enters the U-235 is most favourable to their absorption.

Most materials investigated show a marked tendency to absorb neutrons, a characteristic that disqualifies them for structural uses in reactors. This is dramatically demonstrated by the use of cadmium in control devices. Cadmium has such an affinity for neutrons that a very small amount stops completely the operation of a reactor. Cadmium absorbs enough neutrons to reduce the ratio of the number of neutrons available for fission, to the number of fissions required to maintain operation to less than one. When this occurs action ceases, since a single neutron can produce but one fission. Obviously such a material is useless as a structural material in a reactor. Unfortunately, the same is true of the common structural materials such as steel. This also explains why the materials must be of unusually high purity. The existence of impurities to the extent of a few parts per million in materials in which neutron absorption is relatively low makes these materials unsuitable for use in a reactor.

Similarly the fact that the fission products act as neutron absorbers explains the necessity for removal of these products from time to time during continued operation of a reactor. As more and more of these products are formed, an increasing percentage of the available neutrons is absorbed. This requires a reduction in the amount of control material in the reactor. Ultimately there comes a time when this process cannot be continued, i.e., when all of the control material has been removed. Unless means are provided to remove "poisoned" charges during operation, the reactor must thereupon be shut down and the fuel charge replaced. In any power generating application it seems essential, or at least highly desirable, that continuous operation be possible.

From considerations such as the foregoing, one type of engineering problem appears. Once complete information is obtained as to the neutron-absorption characteristics of materials it will then be necessary to obtain the structural characteristics of those that appear applicable in reactors. This is an engineering problem that can and will be handled by engineers.

Another element that will require engineering consideration arises from the fact that the reactor will in all probability be required to operate at high temperatures, where heat engines are more efficient. This requirement creates additional problems for the engineer. For example, questions will arise as to the corrosion of materials subjected to high temperatures. Probably data must yet be obtained relative to the corrosive effect of various gases or liquids used as coolants or heat transfer media in contact with materials suitable for use within a reactor. These are typically engineering problems.

The high temperatures expected in power reactors introduce additional unknowns that must be determined prior to practical use. The mechanical properties, particularly at high temperatures, of those materials that now appear practical for use in construction of a reactor are largely unknown. Acquisition of the required data is possible by well-known methods but the job is lengthy.

Radiations incident to the process of nuclear fission pose a large number of new problems for the engineer. The need for adequate shielding for operating personnel has been mentioned many times. This problem is more complex, however, than may be first realized. Consider, for example, the probable electric-power generating system. Heat generated in the reactor is transferred by a suitable gas to a heat exchanger in which water is boiled to produce steam for use with a conventional turbine generator. The possibility of leaks through which radioactive material might pass poses a serious problem. At first glance it might be assumed that the only radiation problem is that concerned with the reactor. However, this is not the case. The gas coolant itself may become radioactive and in addition probably would gather other radio-active substances such as dust from the reactor. Presumably the heat-transfer unit in which steam is formed is such as to prevent transfer of radioactivity to the steam by direct radiations. However, a leak would permit radioactive material to contaminate the steam. Depending on the magnitude of this effect, it might be necessary to shield the turbine and piping also. Obviously this is undesirable, therefore much engineering effort would be expended in assuring the absence of leaks.

Continuous operation of a nuclear power-generating equipment is desirable if not essential. This means that the "fuel" must be replaced at intervals during operation. Because of the intense radiation within the reactor, extraction of the "ash" laden nuclear fuel can be accomplished only by remote control. The detail problems connected with the mechanical design of such devices are determined by the particular requirements of the reactor with which they would be associated. However, several general types of problems arise. Presumably the fuel will be distributed throughout the reactor, therefore the device must be so designed as to permit selection, removal, and replacement of fuel elements in any section. The device itself will be continuously bombarded by radiations so that it in turn becomes radioactive. This suggests the impracticality of servicing the device once it has been placed in operation.

Factors such as the effect of radiation on lubricants are at present unknown. These must be determined prior to the use of lubricants with the equipment. Also, assuming that it will be found desirable to have electrical elements in the fuel-handling equipment, their reaction to radiations must be fully explored. For example, it would be fatal to use equipment in which the electrical insulation deteriorates under long-continued radiation bombardment to the point of failure.

Another matter is important, although little has been said regarding it. For many years radiations such as X-rays have been known to affect the properties of certain materials. For example, crystals of potassium chloride change colour under X-ray bombardment but return to normal upon removal of the radiation. Radiations within a nuclear reactor are many times more intense than any previously available. The possible effect of such radiations upon the crystal lattice of materials used in a reactor when exposed continually for long periods of time is yet to be determined. This problem rests squarely to-day in the hands of the research worker.

The foregoing has touched on some of the problems that become apparent when one considers the application of atomic energy to generation of electric-power. Many will require additional knowledge, to be obtained by fundamental research work, before the engineer is able to tackle the design problems to be expected.

The speed with which these and other problems are solved depends, obviously, in large measure on the number of men that can be applied to the work. Most of the fundamental work will have to be done by nuclear physicists and chemists working in government and industrial research laboratories. However, as these men progress with their work many problems will arise that can be turned over to engineers and scientists not necessarily specialists in the field of nuclear physics. Members of the Atomic Energy Commission have recently stated their realization of this possibility and propose to take advantage of it. This is indeed encouraging. If the limited pool of nuclear physicists and chemists now available had to do all the work necessary for the practical application of nuclear energy in the electric power field, many years would pass before the first demonstration could be made. However, with a basic plan that envisions tapping the much larger supply of trained engineers and scientists available in industry and applying them to work on appropriate problems, it seems certain that the advent of nuclear energy in the electric power field is much nearer than otherwise.

As it stands to-day, comparatively little of man's store of knowledge is applicable toward hastening the day of controlled atomic energy. Handbooks and manuals chart and tabulate physical and chemical properties of the many available materials, metallic and non-metallic, under a variety of conditions of temperature, humidity, pressure, solvents and so forth—but not under those expected in an atomic reactor. The immense task of measuring, recording, and tabulating these properties with the additional variables of atomic conditions, may have to be done all over again; for some materials and some properties, it certainly shall.

There is indeed a big job to be done, so big, in fact, that one could not be blamed for being pessimistic about the possibility of a successful outcome. Perhaps the most optimistic note to be found lies in the fact that atomic energy is still a young science, not yet half a century old, and that knowledge of it is still meagre.

Perhaps, instead of being pessimistic, an "ignorance is bliss" attitude would be better. Nevertheless, the realization of the magnitude of the job and the steps being taken to put adequate manpower to work on it give assurance that the necessary research and development will be done and done in as short a time as can reasonably be expected.

RUSSIA

An Accurate Method of Calculation for the Operation of Steam and Air Hammers

By W. F. SHTEGLOV. (From *Vestnik Inzhenerov i Tehnikov*, No. 9-10, 1946, pp. 286-291, 7 illustrations.)

THIS method of calculation is based on indicator diagrams constructed according to the equation

$$\frac{dp}{dH} = k\left(\frac{\varphi f \omega_0}{Fc} - 1\right)\frac{p}{H} \quad .. \quad .. \quad (1)$$

where $\frac{dp}{dH}$ is the tangent of α (see Fig. 1), k is the adiabatic coefficient, φ is the coefficient of consumption, f is a variable in cm² denoting valve area permitting steam (air) flow in either direction, ω_0 is the theoretical velocity of steam (air) flow into the cylinder, F is the effective piston area in cm², c is the piston velocity in m/sec, H is the piston stroke in m taking into account the unswept volume, and p is the cylinder pressure in atmospheres absolute.

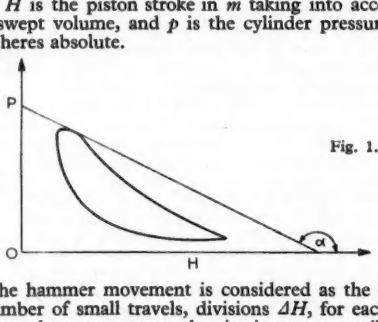


Fig. 1.

If the hammer movement is considered as the sum of a number of small travels, divisions ΔH, for each of which p and c are constant, the rise in pressure will be

$$\Delta p = k\left(\frac{\varphi f \omega_0}{Fc} - 1\right)\frac{p\Delta H}{H} \quad .. \quad (2)$$

and aided by this formula the indicator diagram can be determined.

The values of c in eq. (2) may be calculated from the formula for motion with constant acceleration, for each division of the cylinder, and from the graphs of speed over cylinder length an auxiliary curve $1/c$ can be drawn to give the time by integrating it graphically. An examination of eq. (1) will show which factors determine the pressure curves in indicator diagrams. The upward movement of the piston in the bottom half and the downward movement in the top half of the cylinder are regarded as being positive.

UPWARD MOVEMENT.

(a) BOTTOM HALF OF CYLINDER.

The pressure in the cylinder at the time of steam inlet is not always lower than the pressure in the live steam piping p_0. This condition sometimes occurs in connexion with press hammers when during the movement of the hammer downward, a "cushion" effect is obtained in the bottom half of the cylinder. If $p_0 > p$, steam (air) will enter the cylinder leading to $\omega_0 > 0$. In this case, the character of the pressure curve will depend on the ratio of absolute values of steam consumption $\varphi f \omega_0$ and the volume covered by the piston in each second Fc. When the hammer starts moving, the velocity of the piston is small and $\varphi f \omega_0 > Fc$, which means that $dp/dH > 0$ and the pressure is rising, but as the movement continues velocity c rises and the value of f decreases. When the quantity of steam per second becomes smaller than the volume swept by the piston i.e. $\varphi f \omega_0 < Fc$ the pressure starts to fall. When steam (air) is initially exhausting from the cylinder, i.e. $p_0 < p$, $\omega_0 < 0$ and $dp/dH < 0$ in all cases and the pressure in the cylinder will be falling.

For the divisions corresponding to the exhaust connexion, p_e in the exhaust piping will always be less than p in the cylinder and the pressure will fall.

(b) TOP HALF OF CYLINDER.

When the piston moves into the exhaust port belt steam will be exhausted, because $p > p_e$ and $\omega_0 > 0$. The pressure curve will depend on the ratio of absolute

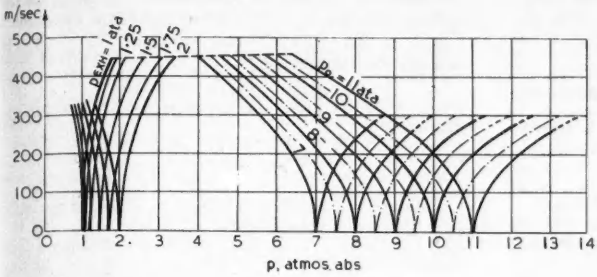
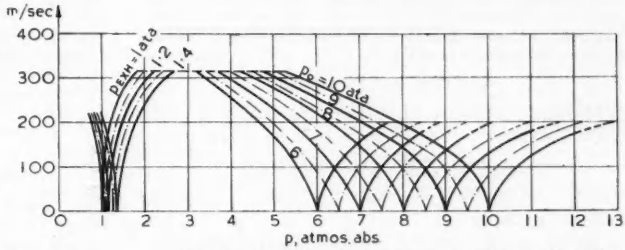


Fig. 2 (left). **Velocity of steam in the minimum division of the stroke as a function of the pressure in the cylinder.**

Fig. 3 (below). **Velocity of air in the minimum division of the stroke as a function of the pressure in the cylinder.**



values of steam (air) quantity to the volume swept by the piston. If $\varphi f \omega_0 > Fc$ the pressure rises. Along the upper inlet port belt normally $p_0 > p$ and dp/dH will be negative so that the pressure will rise.

DOWNWARD MOVEMENT.

(a) Top Half of Cylinder.

Observations made of the upward movement of the piston in the bottom half of the cylinder correspond to the downward movement of the piston in the upper half except that the condition of $p_0 < p$ does not occur in the inlet port belt. The pressure, therefore, depends entirely on the ratio $\varphi f \omega_0/Fc$ and either rises or falls. In the exhaust port belt pressure always falls.

(b) Bottom Half of Cylinder.

The observations made for the movement of the hammer upwards for the top half, are relevant in this case. In the exhaust port belt the pressure falls, if $\varphi f \omega_0 > Fc$ and rises if $\varphi f \omega_0 < Fc$.

In the belt preceding the inlet ports the pressure rises as the hammer moves down. In view of the high velocities of the hammer through this portion of the travel the pressure rises more rapidly than the corresponding pressure in the upper half. The point where $p = p_0$ represents the turning point of the indicator diagram, which becomes convex instead of concave. Pressure rises until $\varphi f \omega_0 < Fc$ and this pressure rise leads to a higher exhaust velocity ω_0 and consequently, even if $\varphi f \omega_0 > Fc$, the pressure curve may fall while remaining convex. The indicator diagram will show a loop, which frequently occurs in practice, when there is a high back pressure exerted by the steam (air) in the bottom.

The observations made here deal with typical cases for the inlet and exhaust port belts and the belt preceding the inlet ports, and the pressure curves during compression and expansion will correspond to the adiabatic curves with a coefficient k. In fact, however, there is some deviation from the adiabatic curves due to steam leakage and heat exchange.

The various terms of eq. (2) have yet to be determined. The values of k are well known for various substances, and φ can be determined experimentally from the indicator diagram and eq. (1).

Generally it is found that for die hammers $\varphi = 0{\cdot}15 - 0{\cdot}2$ and for forging hammers $\varphi = 0{\cdot}2 - 0{\cdot}25$.

To find the minimum free flow sections for different points of the indicator diagram it is necessary to ascertain the law governing the degree of opening of the inlet valves.

The theoretical velocity of steam entry and exhaust ω_0 can be obtained from Figs. 2 and 3, which are based on the following well-known thermodynamic formulae

$$\omega_0 = \sqrt{2g \frac{k}{k-1} p_0 v_0 \left[1 - \left(\frac{p}{p_0}\right)^{(k-1)/k}\right]} \text{ m/sec} \quad (3)$$

when $p < p_0$ and p_0 and v_0 are the pressure and specific volume in the live steam main ; and

$$\omega_0 = \sqrt{2g \frac{k}{k-1} p_0 v_0 \left[\left(\frac{p}{p_0}\right)^{(k-1)/k} - 1\right]} \text{ m/sec} \quad (4)$$

when $p > p_0$.

For the exhaust port belt the formulae are similar, p_0 and v_0 being replaced by p_u and v_u.

The above formulae are valid only for sub-critical values, when $p/p_0 > \beta_{cr}$ and $pu/p > \beta_{cr}$; when $p/p_0 < \beta_{cr}$ and $p_u/p < \beta_{cr}$, ω_0 becomes a constant.

The graphs in Figs. 2 and 3 are based on dry saturated steam and dry air at 30 deg. C.

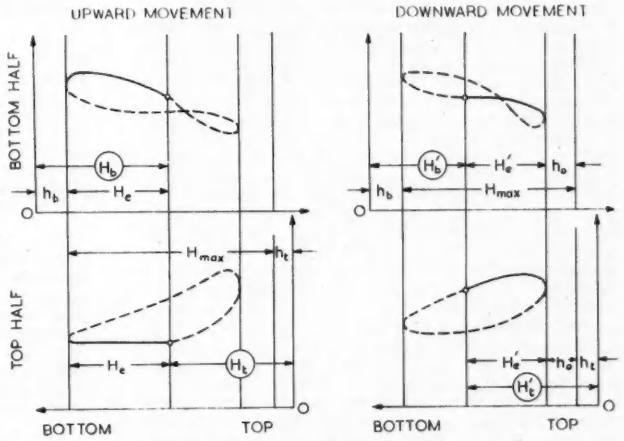


Fig. 4.

The piston velocity for each division of the stroke can be obtained for the movement up and down respectively from the following formulae :—

$$c_n = \sqrt{c^2_{n-1} + 2g\,\Delta H\,[b\,(\alpha\,p_b - p_t + 1 - \alpha) - 1{\cdot}1]} \text{ m/sec} \quad (5)$$

and

$$c_n = \sqrt{c^2_{n-1} + 2g\,\Delta H\,[b\,(p_t - \alpha\,p_b + \alpha - 1) + 0{\cdot}9]} \text{ m/sec} \quad (6)$$

where c_n is the final velocity in a division ΔH ; c_{n-1} is the initial velocity in a division ΔH ; b is the ratio of the top surface of the piston F_t in cm² to the effective weight of the falling parts G_d in kg, $b = F_t/G_d$; and α is the ratio of the effective bottom surface of the piston F_b to its top surface F_t, $\alpha = F_b/F_t = (D^2 - d^2)/D^2$. Friction is assumed as $0{\cdot}1\ G_d$ and the values for H are chosen as shown in Fig. 4.

UPWARD MOVEMENT.

Bottom half cylinder : $H = H_p = H_e + h_b$.

Top half cylinder: $H = H_t = H_{max} + h_b - H_e$.

DOWNWARD MOVEMENT.

Bottom half cylinder: $H = H_b' = H_{max} + h_b - H_e' - h_0$.

Top half cylinder: $H = H_t' = h_t + h_0 + H_e'$

where H_e is the value of effective upward movement for a given point in the indicator diagram, H_e' is the value of effective downward movement for a given point in the indicator diagram, h_t and h_b are the values for top and bottom unswept volume in the cylinder, H_{max} is the maximum possible movement of the hammer, and h_0 is the distance by which the piston does not reach the top dead point.

Thus eq. (2) can be evaluated. The pressure for starting points will be for the top half of the cylinder

$$p_{t0} = 1{\cdot}2\,p_u - 0{\cdot}2 \text{ atm. abs.} \quad (7)$$

and for the bottom half :—

$$p_{t0} = (p_0 + p_s)/2 \text{ atm. abs.} \quad (8)$$

where p_s is the minimum pressure required to lift the hammer.

$$p_s = \frac{F_t(p_t - 1 + \alpha) + 1{\cdot}1\,G_d}{\alpha\,F_t}$$

Initially the divisions H should be small, in the order of 1 mm, and subsequently increasing. Eq. (5) will give the velocity at the end of the first division. The initial velocity c_{n-1} can be assumed as zero for forging hammers and $0{\cdot}2$ — $0{\cdot}4$ m/sec for die hammers. Figs. 2 and 3 will give the velocities ω_0 and eq. (2) will give the rise in pressure. For this purpose $c = (c_{n-1} + c_n)/2$. The pressure at the end of the first division will be $p_1 = p_0 + \Delta p_1$, and with p_1 as the initial pressure of the second sector, p_2 can be obtained similarly. The length of divisions can be selected depending on the steepness of the pressure rise and ΔH must be selected so that Δp does not exceed $0{\cdot}3$ — $0{\cdot}4$ atm. abs.

In order to shorten the calculations a set of nomograms may be prepared.*

By means of the indicator diagrams and curves for the velocity of the hammer found thus, the characteristic coefficients such as speed of impact, energy of impact, and number of impacts per minute, can be obtained.

The comparison of results obtained experimentally and analytically, gives valuable clues to the condition of hammers and, obviously, the method described here can also be used for their design.

*The construction of such nomograms is explained and illustrated in the original paper.—Ed., E.D.

FRANCE

Influence of Production Factors on the Magnetic Properties of Transformer Sheet

By G. DELBART, R. POTASZKIN, and M. SAGE. (From *Revue de Métallurgie*, Vol. 44, No. 7/8, July/August, 1947, pp. 193-209, 21 illustrations.)

THE magnetic properties of transformer sheet depend on so many interdependent production factors that it is difficult to isolate the individual effect of any of them. For this reason very few results of systematic investigations are known although they could lead to direct practical improvements. However, a few results of investigations under full manufacturing conditions are given here. Electric losses are due to eddy currents, produced by variations of magnetic induction, and, to a larger extent, to hysteresis losses attributed to changes of elementary magnetic domains (Weiss domains). Eddy current losses are minimized by the use of thin, insulated iron sheet having a low electric conductivity, and hysteresis losses by eliminating any influence that tends to stiffen the magnetic domains, i.e. by avoiding any impurities that do not go into solution with the iron, too high a carbon content, gas inclusions, a cold worked irregular grain structure, and by producing a coarse grained iron. A high silicon content raises the transformation point of iron and makes annealing possible at higher temperatures so that a coarser grain is obtained. It also reduces the electric conductivity, increases the solubility of carbon, and de-oxidizes the iron, thereby appreciably reducing magnetic losses.

In the tests described, all the stages of manufacture, from the furnace to the finished sheet, were strictly controlled and the most important production factors were varied as widely as possible, viz., the composition and production of iron, ingot weight, number of heats during rolling into slabs, thickness of slabs, and the conditions during rolling into sheet, including the starting and finishing temperatures.

A. Composition of Iron. Soft iron with high silicon content (3 to 4 per cent) was used. Two main types of iron could be distinguished, as those with completely dissolved carbon and those with some carbon out of solution. Average results were better i.e. the losses were smaller, with the first group. However, since complete dissolution of carbon can only be attained by an increase in the silicon and aluminium content, the iron of the first group was rather brittle and led to a considerable number of rejects.

B. Rolling into Slabs. In both cases, rolling at one heat only seems to lead to the best results. The optimum slab thickness is, in the case of iron with completely dissolved carbon, about $\frac{1}{2}$ to $\frac{5}{8}$ inch; iron losses are increased if slabs are chosen thicker or thinner. For iron with carbon out of solution, the slab thickness is of little influence, although, on an average, thin slabs below $\frac{5}{8}$ inch thickness give better results. Finishing temperature should not be below 900 deg. C., or not much below, if rolling is to be done at one heat.

C. Rolling into Sheets. This was done with not less than three heats. For best values (losses below 0·55 Watt

per lb.), four heats should be arranged. Finishing temperature should not be below 750 deg. C. The number of passes during the last heat should be as small as possible, two only if possible, so that the finishing temperature can be very high.

D. Annealing. Certain sheets showed, even after annealing at 850 deg. C. in a controlled atmosphere, a slightly elongated and irregular grain structure. A second annealing at 900 deg. C. definitely improved those sheets which had not been completely recrystallized during the first annealing. Some of those sheets which had shown a completely annealed micro-structure after the first annealing, could be improved by the second annealing, but others showed no change. An annealing temperature of 850 deg. C. seems sufficient for some, but not high enough for other types of sheet.

D. General Remarks. Sheets made from castings with identical carbon content can have very different magnetic qualities. De-carburization attains appreciable proportions only during the final stages of rolling. The form of the included carbon, either in solution or precipitated, therefore plays an important role during the first stages of deformation. The stage at which the carbon content falls below the solubility limit influences the structure and final qualities of the sheet. If precipitated carbon persists up to the final stages of rolling, the self annealing and recrystallization of the hot rolled sheet is continually inhibited and reduced in its effect, and the final magnetic qualities of the sheet deteriorate accordingly. Deformation at too low a temperature is unfavourable throughout the whole process of manufacture but especially so during the last stages. Obviously, manufacturing techniques different from those on which the tests were based may lead to good results under different conditions. The whole problem has been little dealt with but a start is made here by publication of results which, however, are tentative only and require confirmation by further investigations.

RUSSIA

The Measurement of Residual Stresses by Means of Drilling Holes

By D. G. KOURNOSSOV and M. V. JAKUTOVICH. (From *Zavodskaya Laboratoria*, No. 11-12, 1946, pp. 960-967, 2 illustrations.)

THIS paper covers the general case of measuring residual stresses in a plane, when neither the magnitude nor the direction of the main stresses are known. The method described is based on the measurement of creep, caused by the loss of equilibrium between residual stresses due to drilling a small hole, and on the calculation of stresses made possible by that measurement. The stressed surface layer of a flat body after a hole has been drilled can be considered, with sufficient accuracy, as a plate in which a round hole has been made and, to simplify calculations, it may also be considered to be of infinite dimensions.

The equations for the components of creep in the radial and tangential directions are :—

$$\left.\begin{aligned} U &= \frac{S+K}{2E}\left[(1-\mu)\,r + (1+\mu)\frac{a^2}{r}\right] + \\ &+ \frac{S-K}{2E}\left[(1+\mu)\left(r-\frac{a^4}{r^3}\right) + \frac{4a^2}{r}\right]\cos 2\vartheta \\ V &= \frac{S-K}{2E}\left[-(1+\mu)\left(r+\frac{a^4}{r^3}\right) - \right. \\ &\left. - (1-\mu)\frac{2a^2}{r}\right]\sin 2\vartheta \end{aligned}\right\} \quad (1)$$

where S and K are stresses on the contour of a plate of infinite dimensions; a is the radius of the hole; r and ϑ are polar co-ordinates of the selected point; E is Young's modulus; and μ is Poisson's ratio.

GENERAL EQUATIONS FOR THE CALCULATION OF RESIDUAL STRESSES.

To obtain the magnitude and direction of the main stresses it is necessary to know three deformations and, to simplify the calculation, the creep U in three directions has been selected.

U obtained from eq. (1) is composed of two parts :— the movement U_p caused by a state of stress in a plate without holes, and U_h caused by the change to a state of stress due to the drilling of a hole.

$$U = U_p + U_h.$$

The measured movement is U_h, therefore

$$U_{meas} = U - U_p \quad .. \quad .. \quad (2)$$

U_p is obtained from eq. (1) if $a = 0$

$$U_p = \frac{(S+K)(1-\mu)\,r}{2E} + \frac{(S-K)(1+\mu)\,r\cos 2\vartheta}{2E} \quad .. \quad (3)$$

If eqs. (1) and (3) are introduced into eq. (2)

$$U_{meas} = \frac{S+K}{2E}(1+\mu)\frac{a^2}{r} + \frac{S-K}{2E}\left[\frac{4a^2}{r} - (1+\mu)\frac{a^4}{r^3}\right]\cos 2\vartheta \quad .. \quad (4)$$

The three directions in which the creep is measured are :—

$$\varphi_1 = \vartheta\ ;\ \varphi_2 = \vartheta + 90°\ ;\ \varphi_3 = \vartheta + 45°.$$

where ϑ is the angle between the direction of one of the main stresses and one of the directions in which creep is measured. U from eq. (4) in the three directions will be

$$\left.\begin{aligned} U_{\varphi_1} &= \frac{S+K}{2E}(1+\mu)\frac{a^2}{r} + \frac{S-K}{2E}\times \\ &\times\left[\frac{4a^2}{r} - (1+\mu)\frac{a^4}{r^3}\right]\cos 2\vartheta \\ U_{\varphi_2} &= \frac{S+K}{2E}(1+\mu)\frac{a^2}{r} - \frac{S-K}{2E}\times \\ &\times\left[\frac{4a^2}{r} - (1+\mu)\frac{a^4}{r^3}\right]\cos 2\vartheta \\ U_{\varphi_3} &= \frac{S+K}{2E}(1+\mu)\frac{a^2}{r} - \frac{S-K}{2E}\times \\ &\times\left[\frac{4a^2}{r} - (1+\mu)\frac{a^4}{r^3}\right]\sin 2\vartheta \end{aligned}\right\} \quad (5)$$

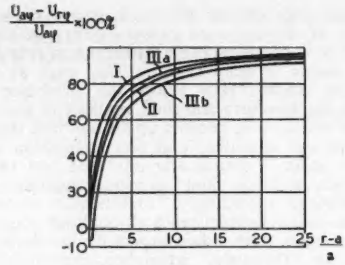


Fig. 1a. **Creep curves.**

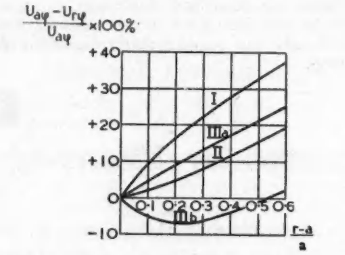


Fig. 1b. **Creep curves drawn to a larger scale.**

(I) $K = S$; $\vartheta = 0$; $\vartheta = 90°$.
(II) $K = -S$; $\vartheta = 0$; $\vartheta = 90°$.
(IIIa) $|S| > 0$; $K = 0$; $\vartheta = 0$.
(IIIb) $|S| > 0$; $K = 0$; $\vartheta = 90°$.

The angle ϑ is derived as follows :—

$$\tan 2\vartheta = \frac{2(U_{\varphi_2} - U_{\varphi_3})}{U_{\varphi_1} - U_{\varphi_3}} + 1 \quad .. \quad (6)$$

After the angle ϑ has been ascertained, the directions of main tensions can be found. These tensions S and K derived from eq. (5) are

$$\left.\begin{aligned} S &= \frac{r\,E\,(U_{\varphi_1} - U_{\varphi_2})}{2a^2(1+\mu)} + \\ &\quad + \frac{E\,(U_{\varphi_1} - U_{\varphi_2})}{2\left[\frac{4a^2}{r} - (1+\mu)\frac{a^4}{r^3}\right]}\cos 2\vartheta \\ K &= \frac{r\,E\,(U_{\varphi_1} - U_{\varphi_3})}{2a^2(1+\mu)} - \\ &\quad - \frac{E\,(U_{\varphi_1} - U_{\varphi_2})}{2\left[\frac{4a^2}{r} - (1+\mu)\frac{a^4}{r^3}\right]}\cos 2\vartheta \end{aligned}\right\} \quad (7)$$

Eqs. (6) and (7) represent the general solution of the problem in a plane, when neither the magnitude nor direction of the two main stresses are known.

Determination of Residual Stresses, when the Directions of the Main Stresses are Known.

This is a special case of the problem often met in practice, and the solution requires the measurement of creep in two directions only. The equations for U, S and K become simpler especially when $r = a$.

Eqs. (5) for points on the contour of the hole and in the directions of the main stresses ($\vartheta = 0°$ and $\vartheta = 90°$) become for $U_{r=a} = U_1$ and $U_{r=a} = U_2$

$$\vartheta = 0 \qquad \vartheta = 90°$$

With $r = a$, $\vartheta = 0$ and $\vartheta = 90°$

$$\left.\begin{aligned} U_1 &= \frac{2aS - (1-\mu)\,aK}{E} \\ U^2 &= \frac{2aK - (1-\mu)\,aS}{E} \end{aligned}\right\} \quad .. \quad .. \quad (8)$$

Eqs. (7) become

$$\left.\begin{aligned} S &= \frac{E\,[2U_1 + (1-\mu)\,U_2]}{a\,(1+\mu)\,(3-\mu)} \\ K &= \frac{E\,[2U_2 + (1-\mu)\,U_1]}{a\,(1+\mu)\,(3-\mu)} \end{aligned}\right\} \quad .. \quad .. \quad (9)$$

When these formulae are used it must be born in mind that, owing to various inaccuracies, the actual creep will differ somewhat and it is necessary to use the correction curves given in Fig. 1, which take into account the reduction of creep, caused by the drilling of a hole, as the distance from the contour of the hole increases. The curves in Fig. 1 are designed for three special cases

$$K = S\,;\; K = -S\,;\; K = 0\,;\; |S| > 0.$$

and indicate that the amount of creep decreases sharply with the distance from the hole and therefore for maximum accuracy, the tensometer must be placed as near as possible to the hole.

If it is desired to use the simplified formulae (9) instead of (7) the corrected values for U_1 and U_2 must be introduced, otherwise formulae (7) must be used.

The Depth of the Drilled Hole.

The change in the stress of the area of the body under examination is brought about by the drilling of a small hole and causes a deformation, and the question raised is regarding the depth to which the hole can be drilled without causing a deformation of the surface. Experiments with a steel plate of 35·8 × 104 × 800 mm. showed that the deformation obtained by drilling to a depth of 1·5 to 2·0 times the diameter is of

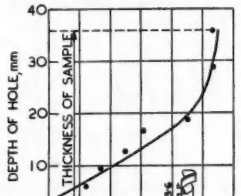


Fig. 2. **Curve showing creep as a function of the depth of the hole.**

the same order as when the hole is drilled right through, see Fig. 2. Creep was measured with the aid of a string type tensometer.

Limitation of the Method.

(1) The drilling of a hole causes higher stresses which must not exceed the yield point.
(2) If *Kirsch's* equation is to be used the zone of influence of the hole must have main stresses which are of the same order. This limits the zone to a radius of $20a$, outside which the hole becomes practically ineffective.

(3) The application of *Kirsch's* equation presupposes a uniform stress across the sample piece.

(4) A condition of the method is the possibility of drilling a hole of 8-10 mm. diameter and 16-20 mm. deep.

Measurement of Residual Stresses.

Experiments were carried out with the aid of this method on the rolls of hot rolling mills, some of which were rejected owing to residual stresses. One experiment was carried out on a roll which had broken into two approximately equal parts. Measurements were carried out on the side surface of one of these parts having the following dimensions

L_r = 620 mm. length of barrel;
D_r = 560 mm. diameter;
L_n = 740 mm. length of roll neck;
D_n = 400 mm. diameter of roll neck.

In this case the direction of main stresses was known and it was only necessary to measure the creep in those directions. The creep was measured with the aid of string tensometers with a 60 mm. base.

$$\Delta l = 329 \times 10^{-7} N^2 \quad .. \quad .. \quad (10)$$

where Δl is the absolute deformation of the string in microns ; N is the natural frequency of the string oscillations.

The tensometers were placed in the direction of the roll axis and at right angles to it.

The accuracy of the method increases with the diameter of the hole, but for practical purposes 8 to 10 mm. is sufficient, and gives an error of less than 5 per cent.

S and K are axial and tangential stresses.

T_5 means the tensometer in the axial direction.

T_6 means the tensometer at right angles to T_5.

a is the radius of the hole.

h is the depth of the hole.

Δl_0 and Δl_k are the absolute elongation of the string before and after the drilling of a hole.

Δl is the measured change of the tensometer base caused by the drilling of the hole.

Δl_1 is the actual change of the base.

U_1 and U_2 are the creep along the contour of the hole in the direction of the main stresses.

η is the difference of the creep of points along the contour and of points of support for the tensometer, divided by the creep along the contour.

Δr_5 and Δr_6 are the distances of the front legs of T_5 and T_6 from the contour.

$\Delta r_5'$ and $\Delta r_6'$ are the distance of the rear legs of T_5 and T_6 from the contour.

Owing to its particular construction the creep measured by the tensometer is 15 per cent smaller than the true creep.

$$\Delta l_1 = \Delta l + 0{\cdot}15\,\Delta l = 1{\cdot}15\,\Delta l \quad .. \quad (11)$$

Table I gives the results measured.

If tensometer T_5 is considered, the absolute deformation of the tensometer string before and after the hole is drilled, is in accordance with eq. (10) :—

Table I.

Tensometer marking	h mm	Readings in generator divisions	N in cycles
T_5	0	16 — 32	1061
	13	15 — 22	1132
T_6	0	9 — 23	1614
	13	8 — 78	1646

a = 4 mm, Δr_5 = 1·5 mm, $\Delta r_5'$ = 61·5 mm,
Δr_6 = 1·8 mm, $\Delta r_6'$ = 61·8 mm.

$\Delta l_0 = 329 \times 10^{-7} \times 1061^2 = 36{\cdot}8\,\mu$;
$\Delta l_k = 329 \times 10^{-7} \times 1132^2 = 41{\cdot}9\,\mu$;
$\Delta l = \Delta l_0 - \Delta l_K = -5{\cdot}1\,\mu$;
$\Delta l_1 = 5{\cdot}9\,\mu$.

The stresses are calculated from the simplified eq. (9) which requires U_1. Fig. 1 gives the correction factor η, the curve for $S = K$ being used.

$$\frac{\Delta r_5}{a} = \frac{1{\cdot}5}{4} = 0{\cdot}38 \qquad \frac{\Delta r_5'}{a} = \frac{61{\cdot}5}{4} = 15{\cdot}4$$

$$\eta = 27{\cdot}5 + 6 = 33{\cdot}5 \text{ per cent.}$$

From this

$$U_1 = \frac{-5{\cdot}9 \times 100}{66{\cdot}5} = -8{\cdot}9\,\mu.$$

Similarly the tensometer T_6 gives

$$U_2 = -6{\cdot}2\,\mu.$$

With the aid of eq. (9) the stresses can now be determined

$$S = \frac{-2 \times 10^4 \times 22{\cdot}1 \times 10^{-3}}{14} = -31{\cdot}6 \text{ kg/mm}^2$$

$$K = \frac{-2 \times 10^4 \times 18{\cdot}6 \times 10^{-3}}{14} = -26{\cdot}6 \text{ kg/mm}^2.$$

For the above calculation it was assumed that

$$E = 2 \times 10^4 \text{ kg/mm}^2, \text{ and } \mu = 0{\cdot}3.$$

The results of these experiments are shown in Table II. They prove the existence of high residual compression stresses in the surface layer of the roll. The tensile stresses near the centre of the roll, which are in equilibrium with the compression stresses on the periphery are also high. The axial stresses being higher than the tangential stresses caused the roll to break.

Summary.

(1) The method as described gives an accuracy of ± 5 per cent, which is satisfactory for all practical purposes on large pieces such as forgings, castings, turbine shafts, and large tubes.

(2) The method can be applied to parts of large structures such as bridges without the necessity to dismantle them.

(3) The instruments developed permit the obtaining of results within an hour, and they are therefore suitable for industrial use.

Table II.

Tensometer Marking	h mm	Generator Reading	N Cycles	$N^2 \times$ $\times 10^{-5}$	Δl_0 μ	Δl_k μ	Δl μ	Δl_1 μ	η %	U_1 and U_2	S and K kg/mm²
T_5	0	16 — 32	1061	1126	36·8						
	13	15 — 22	1132	1281		41·9	— 5·1	— 5·9	33·5	— 8·9	— 31·6
T_6	0	9 — 23	1614	2605	85·3						
	13	8 — 78	1646	2709		88·7	— 3·4	— 3·9	37	— 6·2	— 26·6

ITALY

The Economic Dimensioning of Small Transformers

By S. Silleni. (From *Tecnica Italiana*, Vol. 11, No. 5/6, September/December, 1947, pp. 231-235, 2 illustrations.)

1. Introduction.

The dimensioning of a transformer constitutes an indeterminate problem and from the infinite number of possibilities, the solution chosen is the one which offers the best economic advantages. In the case of large transformers, the solution sought is the one which ensures minimum losses and maximum efficiency ; but in the case of small transformers intended for wireless apparatus and similar applications, production cost, and weight and size, are of primary importance. This article, therefore, describes a rapid method for the dimensioning, on the basis of minimum cost, weight, or size, of small naturally cooled single-phase and three-phase transformers.

2. General.

The choice of the current density in the winding and of the induction B_{max} in the iron is determined by the admissible increase of temperature and by the admissible saturation. In the case under consideration, the current density will be assumed to have a given value.

Throughout this article the index 1 is applied to specific values referring to a gross cross-section. For instance u_1 is the current density referred to the total cross-section reserved for the winding, and may be $^1/_2$ to $^1/_5$ or even less, of the actual density if referred to the copper conductors only. It is assumed that the primary and secondary circuits carry the same load in VA and in this case the cross-section available for the winding (the window) is shared in equal parts by the primary and secondary windings.

3. Single-Phase Transformers with Cores of Square Cross-Section.

The square cross-section of the core is adopted because it allows certain advantages to be obtained in the manufacture of the winding. In the case of stampings as indicated in Fig. 1, a window of the dimensions

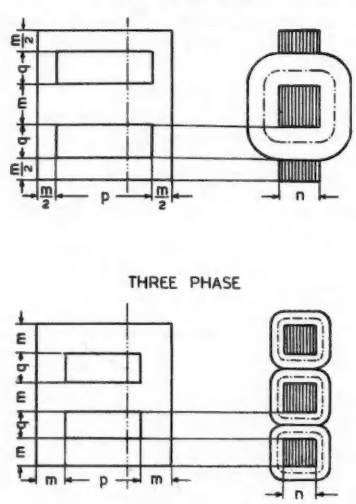


Fig. 1.

$p \times q$ is reserved for the winding. Therefore, the total current flowing in one of the two windings is

$$I = u_1 \frac{F_{Cu}}{2} = u_1 \frac{pq}{2} \quad \text{[amp., sq. m., m.]} \quad .. \quad (1)$$

The e.m.f. induced in one turn :—

$$E = 4{\cdot}44\, B_1 f F_{Fe} =$$
$$= 4{\cdot}44\, B_1 f m n \quad \text{[volt, weber/sq. m., c./sec., m.]} \quad (2)$$

The capacity in VA is, therefore :

$$VA = EI = 4{\cdot}44\, B_1 f m n u_1 p q/2 =$$
$$= 2{\cdot}22\, B_1 f u_1 F_{Cu} F_{Fe} \quad .. \quad (3)$$

Thus :—

$$\frac{F_{Fe}}{F_{Cu}} = x^4 = \frac{m n}{p q} \quad .. \quad .. \quad (4)$$

that is the ratio of the cross-section of the core to the cross-section of the winding, and

$$\frac{p}{q} = k^2 \quad .. \quad .. \quad .. \quad (5)$$

$$VA = 2{\cdot}22\, B_1 f u_1 m^2 n^2/x^4 \quad .. \quad .. \quad (6)$$

Therefore the cross-section of the iron is

$$F_{Fe} = mn = \sqrt{\frac{VA}{2{\cdot}22\ B_1\ f\ u_1}}\ x^2 = C^2 x^2 \quad .. \quad (7)$$

which is a function of a constant C which depends on the capacity in VA of the transformer and other parameters, and of the one variable x.

The cost L' of the transformer is equal to the product of the volume of the core v_{Fe} and the cost of the iron S_{Fe} plus the volume of the winding multiplied by the cost of the material S_{Cu} :—

$$L' = v_{Fe} S_{Fe} + v_{Cu} S_{Cu} \quad .. \quad .. \quad (8)$$

or $L = L'/S_{Fe} = v_{Fe} + S_{Cu} v_{Cu}/S_{Fe} = v_{Fe} + \nu v_{Cu}$ (9)

with $\nu = S_{Cu}/S_{Fe}$ being the ratio of copper cost to iron cost.

For a core with a square cross-section ($m = n$), the following formula can be derived from the foregoing (see also Fig. 1).

$$\frac{L}{C^3} = 2x^3 + 2x \frac{1 + k^2}{k} + \nu \frac{4}{x} + \frac{\nu \pi}{k x^3} \quad (10)$$

to give the cost of the transformer as a function of a single variable x.

From this and the minimum condition

$$6x^6 + 2x^4 \frac{1 + k^2}{k} - 4\nu x^2 - \frac{3\pi\nu}{k} = 0 \quad .. \quad (11)$$

Fig. 2 gives the values for x^2 as a function of ν and of k^2.

Thus, the cross-section of the iron can be calculated directly from eq. (7) and the complete calculation can be carried out graphically by using the nomogram shown in Fig. 4.

4. Three-Phase Transformers.

The calculation of a three-phase transformer as shown in Fig. 1 is entirely analogous to that for single phase transformers.

5. Determination of Transformer Dimensions.

The calculations in paragraph 3 were based on the assumption that it was necessary to use available

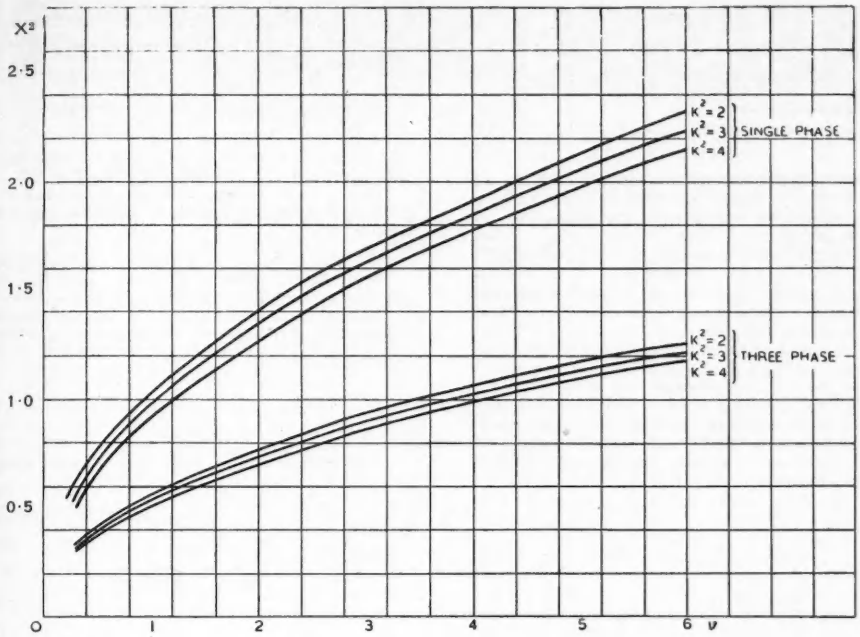


Fig. 2. **Square root of the ratio of the cross-section of core and winding, as a function of the ratio ν of the cost of core and winding ; parameter k^2 is the ratio of length and width of the window (for square cores).**

stamped laminations, with a given ratio k^2 of the width and length of the window, and that the cross-section of the core was to be square.

In the following the method is amplified to determine the optimum values for these two items.

It is obvious, of course, that it is not convenient to reduce the thickness of the packet of laminations below the value for the square cross-section, as this would increase both the amount of iron and copper.

For constant capacity eqs. (2) and (3) can be simplified to

$$F_{Cu}\, F_{Fe} = m\, n\, p\, q = 1 \quad .. \quad .. \quad (12)$$

The following variables are introduced :—

$$y^2 = \frac{p}{q}, z^2 = \frac{n}{m} \text{ and from this and with eq. (12)}$$

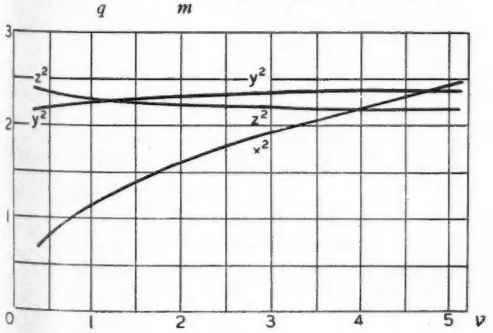


Fig. 3. $x^2 = \sqrt{\frac{F_{Fe}}{F_{Cu}}}$, $y^2 = \frac{p}{q}$ and $z^2 = \frac{n}{m}$ as functions of the ratio of the costs of core and winding ν (rectangular cross-section of core).

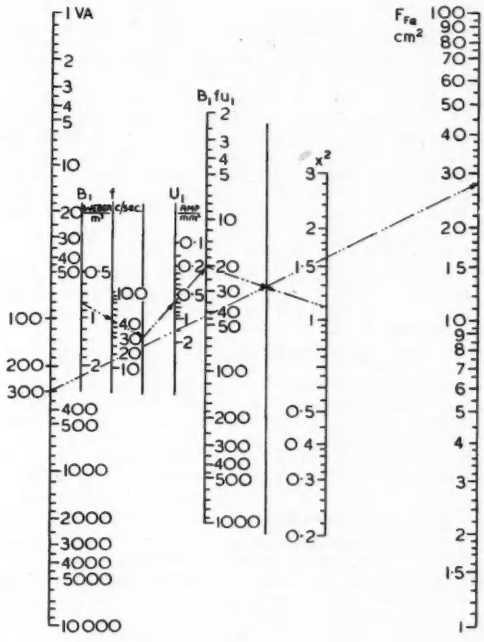


Fig. 4. **Nomogram for calculating the cross-section of the core of a transformer from the known capacity, frequency f, (gross) induction B, (gross) current density u. and the ratio x^2 read from Figs. 2 or 3.**

$$m = \frac{x}{z}\ ;\ n = x\,z\ ;\ p = \frac{y}{x}\ ;\ q = \frac{1}{xy}.$$

The total cost as a function of the sum of the volumes of the core and the winding, and the ratio of the respective costs is again :—

$$L = v_{Fe} + \nu\, v_{Cu} = 2\,m\,n\,(p + q + m) + \\ + \nu\, p\, q\,(2\,m + 2\,n + \pi\, q) \quad (13)$$

By partial differentiation of L with respect to the three variables x, y, and z the three minimum conditions are obtained as functions of the ratio ν of the costs of iron and copper. This is given in Fig. 3 which shows that both y^2 and z^2 deviate very little from 2·25. Yet it would be unwise to assume that the relative dimensions of transformers should be standardized. In fact, the current density having been assumed constant, means in practice that the latter can be still further increased (and with it the capacity of the transformer) if the cooling surface is made larger. From this it follows that, to a minor degree, the capacity of the transformer can be increased if n is somewhat decreased below, and m is increased above, the values corresponding to those calculated for z^2 ; and this argument incidentally justifies the wide-spread choice of a square cross-section for cores.

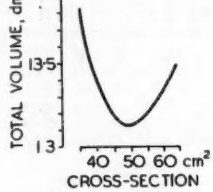


Fig. 5. Total volume of a transformer of given capacity as a function of the cross-section of the core. The choice of the cross-section of the core is not critical for obtaining maximum economy.

In no case is there any justification for choosing the thickness of the packet of laminations to be more than 2·25 times the width of the core, or the length of the window to be less than 2·25 times its width. These values are not critical, however, as can be seen from Fig. 5.

6. The Ratio ν of Copper and Iron Costs.

The choice of the copper and iron costs ratio ν is, in practice, simpler than it might appear at first glance, and a few examples for various conditions are given in the following :—

(a) *Minimum volume* (space) : $\nu = 1$ (see eq. (9)) ;

(b) *Minimum weight* : in this case the term "unit cost of the winding" means the weight per unit of volume, (i.e. the specific gravity of copper γ_{Cu} multiplied by the fill factor f_{Cu}) and the "unit cost of the core" means the weight per unit of volume (i.e. the specific gravity of the iron γ_{Fe} multiplied by the fill factor f_{Fe} of the iron) ; for example, with $\gamma_{Cu} = 8900$, $\gamma_{Fe} = 7600$, $f_{Cu} = 0{\cdot}3$, and $f_{Fe} = 0{\cdot}9$, it is found that $\nu = 0{\cdot}4$.

(c) *Minimum manufacturing cost* : The cost of the unit of volume of the winding (or of the core) is determined by the sum of the market price of the material and the production costs. If a more accurate analysis is required, the prospective value of the material after the economic life of the transformer is terminated, can be deducted from the initial cost. It is

$$\nu = \frac{S_{Cu}}{S_{Fe}} = \frac{S_{Cu\ kg}\ \gamma_{Cu}\ f_{Cu}}{S_{Fe\ kg}\ \gamma_{Fe}\ f_{Fe}} \quad .. \quad .. \quad (14)$$

with a copper price of $S_{Cu\ kg} = 800$ Lire per kg, and an iron price of $S_{Fe\ kg} = 400$ Lire per kg, $\nu = 0{\cdot}8$.

For a more compact construction and a higher price for copper, the value of ν increases and may even exceed unity.

(d) *Minimum operating cost* : The operating costs are proportional to the total losses ; therefore the value of ν equals the value of the losses per unit of volume.

With u the current density in the net copper cross-section and ρ the specific resistance of the conductor, the copper losses per unit of net volume are

$$P_{Cu}' = \rho u^2 \quad .. \quad .. \quad .. \quad .. \quad (15)$$

and per unit of gross volume

$$P_{Cu}'' = \rho u^2 f_{Cu} \quad .. \quad .. \quad .. \quad (16)$$

The corresponding expression for the iron losses is

$$P_{Fe}'' = P_{Fe}'\ \gamma_{Fe}\ f_{Fe} \quad .. \quad .. \quad (17)$$

where P'_{Fe} = iron losses per kg. at a frequency f and the induction B_{max}.
Therefore

$$\frac{P_{Cu}''}{P_{Fe}''} = \frac{\rho u^2 f_{Cu}}{P_{Fe}'\ \gamma_{Fe}\ f_{Fe}} \quad .. \quad .. \quad (18)$$

if the transformer continuously operates on full load. Taking into account the average load through a load factor Θ_{Cu}, and assuming that the specific resistance of the conductor at the operating temperature is $\rho = 20 \times 10^{-9}$ ohm $\times$ metre, then

$$\nu = \frac{\Theta_{Cu}\ u^2\ f_{Cu} \times 20}{P_{Fe}'\ \gamma_{Fe}\ f_{Fe}} \quad .. \quad .. \quad (19)$$

where P_{Fe}' is in watt/kg, γ_{Fe} in kg/dm³, and u in amp./mm² net.

With the aforementioned data, and $u = 2$ amp./mm² net and $P_{Fe}' = 1{\cdot}3$ watt/kg, it is found that $\nu = 2{\cdot}7$.

(e) *Minimum total cost.* Total cost is understood to mean the sum of the manufacturing costs and operating costs capitalized at the time of construction. Without discussing in detail the problems of the economic life of a transformer and of the rate of interest, these two items will be taken into account as a factor T by which the operating cost (per unit of time) must be multiplied to obtain the capitalized value. The operating cost itself is given by the losses multiplied by the price of electricity S_e (lire per joule). Therefore it is

$$\nu = \frac{S_{Cu} + T\ S_e\ P_{Cu}''\ \Theta_{Cu}}{S_{Fe} + T\ S_e\ P_{Fe}''\ \Theta_{Fe}} \quad .. \quad (20)$$

where Θ_{Fe} is the time factor.

Eq. (2) combined with eqs. (16) and (17) gives finally,

$$\nu = \frac{f_{Cu}\ \gamma_{Cu}\ [S_{Cu\ kg} + 19{\cdot}7\ \Theta_{Cu}\ T\ S_e\ u^2}{f_{Fe}\ \gamma_{Fe}\ [S_{Fe\ kg} + 8{\cdot}76\ \Theta_{Fe}\ T\ S_e\ P_{Fe}} \quad .. \quad (21)$$

which is the most general formula for calculating ν.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

CONCRETE

The Choice of Amplitude for Concrete Vibrators.

By N. Y. CHARCHUTA. (From *Mechanizatsia Stroitelstva, Russia*, No. 9, 1947, pp. 10-11, 1 illustration).

THE essential characteristics which determine the satisfactory functioning of concrete vibrators include (a) the weight of vibrating parts (b) the force of vibration (c) the frequency and (d) the area of the vibrator base.

This article deals with the force of vibration for which the amplitude can serve as a measure, and states that other characteristic features have yet to be examined.

The design of a vibrator is determined by the size of the individual particles in the concrete and their cohesive forces. While the vibrator is in operation two kinds of motion take place in the concrete:—

(a) Vibration of the whole mass of concrete.

(b) The relative movement of particles to each other, i.e. the separation of solid particles from the liquid.

The acceleration and velocity of particles is also an important factor for the design of vibrators.

The highest density is obtained when the vibrator and the concrete comprised one single system of oscillation and this is the condition when the force applied and the amplitude of vibration is correctly selected. Excessive amplitude will lead to the vibrator detaching itself from the mass, but insufficient amplitude will not densify the mass.

Experiments proved that for soil the optimum amplitude would be 0·3 + 0·4 mm for frequencies of 1800-3000 cycles/min. For higher frequencies the optimum amplitude would be higher. With concrete the problem is complicated by the considerable adhesive force between the base of the vibrator and the mass.

Experiments have shown that the amplitude of vibration for the mass increases at first and later decreases until it becomes stabilized, for both the vibrator and the mass, towards the end of the compacting process owing to the increased internal resistance of the densified mass. A maximum final amplitude of 0·5 mm was established, but it was found advisable to reduce this to 0·3-0·4 mm as the higher amplitude causes excessive dynamic stresses in the vibrator.

The optimum amplitude is:—

$$P = K_2\left(Q + K_1 \frac{F}{160}\right)$$

where P is the force causing vibration in kg.

Q is the weight of vibrator in kg.

F is the area of vibrator in cm², and

K_1 and K_2 are constants which depend on the proportion of cement to water and the frequency of vibration.

$K_1 = 11$ for concrete with $W/C \geq 0{\cdot}5$.

$K_1 = 15$ for $W/C = 0{\cdot}45$—$0{\cdot}47$.

K_2 varies from 0·5 to 1·4 depending on the ratio W/C and the frequency of vibration.

DISTRICT HEATING

District Heating in the U.S.S.R.

By M. B. PERLIN. (From *Promyshlennaya Energetica, Russia*, No. 11, 1947, pp. 7-8.)

THE first District heating installation was introduced as far back as 1924. During the last pre-war five year plan a great deal of effort was devoted to district heating schemes. The Russian district heating installations differed from those in America in that from the start the generation of power and heat were side by side developments. Extraction type turbines are generally used with an extraction pressure of 1·2 to 2·5 atmos. absolute while hot water under pressure is used for the circulation of heat.

Large scale district heating schemes for cities, of the size of Moscow, presented new problems, such as the economical radius for an installation, temperature regulation in the network, etc. All equipment, including extraction turbines, boilers, etc., were produced in the U.S.S.R.

Most of the new industrial combines and the townships built around them were equipped with their own district heating plants. Before the war the total installed capacity of thermo-electric power stations was 2×10^6 kW and the heat generated was 20×10^6 million calories.

During the war some installations were destroyed but, further east others were built and the total installed capacity remained at its pre-war level.

Since the end of the war the generation of heat by such installations has reached 30×10^6 Mcal per year.

Fuel economy in the generation of electric power amounts to 10 per cent while, for the production of heat, the saving amounts to at least 20 per cent. Thus the total saving of coal in the U.S.S.R. due to thermo-electric stations is over $1{\cdot}5 \times 10^8$ tons. The post-war five year plan caters for a large-scale extension of district heating installations, but in the meantime the U.S.S.R. has more installed capacity in thermo-electric power stations than any other country. Higher temperatures up to 180 deg. C. are planned for distribution of hot water and research is being directed to the question of higher efficiency in the laying of distribution pipework.

ELECTRIC GENERATORS

The Regulation of Load Current of Permanent Magnet Alternators with a Wide Range of Speed Variation.

By S. V. KRAUS and T. G. SOVOKER. (From *Vestnik Electropromyshlennosti, Russia*, No. 9, 1947, pp. 5-8, 6 illustrations).

SMALL synchronous alternators with permanent magnets have found a wide field of application. Special cases are their use on tractors or as wind-driven alternators for the lighting of outlying farms for which the load is constant but the speed of drive varies greatly. To obtain constant voltage at the load terminals a stabiliser is used which keeps it constant in spite of speed and consequent frequency changes, and this is effected by introducing inductances and capacitances into the alternator load circuit which are selected so that the rise in e.m.f. is compensated for by a rise in the total

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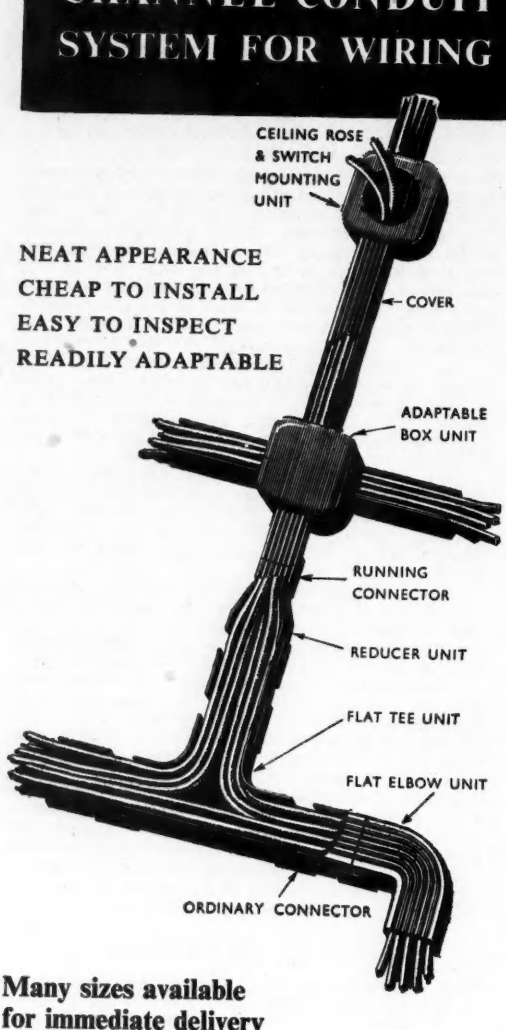

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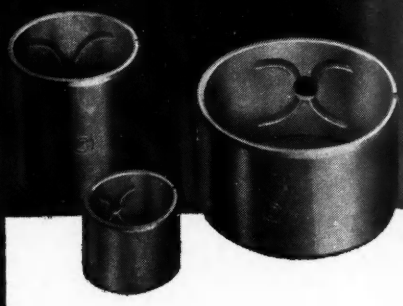

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resistance, as the latter follows the same characteristic as the e.m.f. with increasing speed or frequency. Normally the inductance X_L and capacitance X_C are selected to give $X_L = X_C$ at the lowest alternator speed—which corresponds to the resonance point for the voltage. As the alternator speed increases $X_L - X_C$ or the total resistance grows correspondingly.

In some cases the condensers are not connected into the circuit, but through a current transformer, which allows them to be smaller in size.

A vector diagram will give the following relation:

$$\frac{I}{I_Q} = \frac{f_o^2\sqrt{f_o^2 R_q^2 \times (f_o^2-1)^2}}{f_o^2 R_q + (f_o^2-1)^2 - \Delta f_o^2(f_o^2-1)}$$

where I is the load current

$I_Q = E / x_q$

$x_q = x_{aq} + x_s + x_T$

x_{aq} is the quadrature axis armature reactance.

x_s is the alternator leakage reactance.

x_T is the transformer short circuit reactance.

$R_q = rf / x_q$.

$r = r_a + r_r + r_{Tr} + r_L$.

r_a = the alternator ohmic resistance.

r_T = the transformer ohmic resistance.

r_{Tr} = the transmission line ohmic resistance.

r_L = the load ohmic resistance.

The above equation serves for the construction of a family of curves I / I_Q for a varying ratio of the circuit ohmic resistance to reactance over the frequency and for different values of

$$\frac{x_q - x_d}{x_q} = \Delta$$

$x_d = x_{ad} + x_s + x_T$.

where x_{ad} is the direct axis armature reactance.

Curves thus obtained give the necessary data for designing the stabilisers and it becomes obvious that optimum conditions are obtained for

$\Delta = 0$, i.e. $x_q = x_d$ and $rf = (1 \div 1{\cdot}2)\, x_q$.

Alternators designed on this basis have shown a voltage variation of only 6 per cent for a difference in speed of 77 per cent.

FOUNDRY PRACTICE

Resin Treated Moulding Sands for Grey Iron Castings.

(From *Fonderie, France*, No. 24, December, 1947, pp. 934-938, 4 illustrations).

In a symposium on foundry production technique (21st congress of the French Foundry Association), Mr. Curry, for the American Foundryman's Association, reported on tests and manufacturing experience with resin treated sand. The resin, in the form of a viscous chemical liquid, is mixed with the washed, humid siliceous sand in an ordinary mixer (about 120 gallons per ton of sand) and dried at 150 to 200 deg. C. so that solvent and water evaporate. The solvent can be reclaimed from the vapour. On leaving the drier, the sand grains are unmodified and only covered by a very thin, dark-brown, carbon-rich resin coating which is hard, strongly adherent, resistant to thermal and mechanical stresses, as well as insoluble in water and ordinary solvents. However, it becomes plastic again on contact with the special solvent of the chemical mixture. The coated sand is introduced into the first moulding machine, is mixed there with bentonite, a cereal base binder, the chemical liquid and water, and is thus given the necessary moulding qualities. It can be used in that state or mixed with silica, and can be re-used if bentonite, binder, and chemical liquid are added in sufficient quantities to maintain constant moulding qualities. During moulding, casting, and stripping, the coating will be burnt on some grains, or transferred to others, so that small additions of the chemical in the crushers will suffice to maintain the total amount of coated sand. Synthetic sands normally lose humidity, and therefore plasticity, on their way from crushers to the pattern-plate, but chemically treated sand gains in plasticity, due to the slow plasticizing action of the chemical added in the crushers, and arrives, after 20 to 30 minutes, at the pattern-plate in exactly the right condition. The sand qualities can finally be closely controlled by the kind of binder or by modification of the specifications, without incurring a risk of adhesion due to increased resistance at high temperatures.

The sand has better packing qualities than any other so far used by the author and makes more uniform and denser moulds with less machine ramming, and thus saves time, improves the casting finish and facilitates the maintenance of tolerances. When the coating is completely plastic, the grains lose coherence, and stripping is therefore facilitated. This in turn leads to a better quality of the sand to be re-used, restricts losses, and improves the finish of cast surfaces.

ROLLING MILL PRACTICE

The Speed of Electric Drives for Coilers on Cold Rolling Mills.

By A. N. Iroshnikov. (From *Vestnik Electropromyshlennosti, Russia*, No. 8, 1947, pp. 8-9, 3 illustrations.)

The relation of the speed of the work rolls to that of the coiler in cold strip mills is important because it determines the strip tension which is normally required to be constant. The peripheral speed of the work rolls v is higher than the entrance speed of the strip v_o but lower than the speed v_1 at which the strip leaves the rolls. The increase in speed $s = (v_1 - v)/v$ is explained by the fact that the lowest resistance to deformation is in the direction of the strip movement as the metal slides over the surface of the rolls. The increase in speed depends on the process of plastic deformation, particularly on the draft, and on the value of the strip tension.

Under normal rolling conditions when the strip tension is kept constant, v_1 must be equal to v_c which is the peripheral speed of the coiler:

$$v_c = v_1 = (1 + s)\, v$$

The strip is coiled, after having been rolled, in accordance with the equation for an Archimedian Spiral.

$$R = R_C + \frac{h_1}{2\pi}\varphi$$

h_1 being the thickness of strip, and R_C the radius of the coiler. The differential of the strip length will be $dL = R\, d\varphi$. This, introduced into the previous equation and integrated, gives the length of strip wound on the coiler

$$L = R_C\,\varphi + \frac{h_1}{4\pi}\varphi^2$$

The speed v_C is obtained by differentiating the last equation

$$v_C = \left(R_C + \frac{h_1}{2\pi}\varphi\right)\omega$$

from which

$$n = \frac{60\,(1+s)\,v}{2\pi R_C + h_1\varphi}$$

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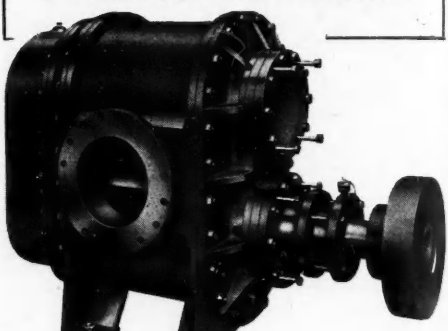

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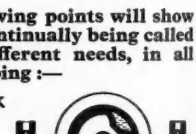

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under constant normal working conditions, i.e. when the starting up process is completed, v_{const} will be the speed of rolling and R_m will be the radius of the coil at that moment, while angle φ will be expressed by the speed of the coiler. With this, and after some transformation, the following equation is obtained for the coiler speed:—

$$n = \frac{30\,R_m}{h_1\,t}\left[t\sqrt{1+\frac{2h_1\,(1+s)\,v_{const}}{\pi\,R^2{}_m}}-1\right]$$

The electric drive for the coiler must be designed to follow a similar falling speed characteristic in order to keep strip tension T constant. To obtain this, the driving motor characteristic must follow a line of constant output

$$N = k_1\,Mn = k_2\,Tv_c = \text{constant}$$

If the voltage at the motor terminals is kept constant, the current must be controlled to give

$$N = k_3\,UI = \text{constant.}$$

SCIENTIFIC INSTRUMENTS AND PRECISION TOOLS

The Proficorder—An Instrument for Recording Waviness and other Surface Profiles.

By E. J. Abbott, E. Rupke, and A. Arbor. (From *Transactions of the ASME*, Vol. 70, No. 4, May, 1948, pp. 263-270, 13 illustrations.)

The instrument described represents a major addition to industrially available surface-measuring instruments. It provides data which are highly essential in the study of surfaces and the processes by which they are obtained. It is unique among such instruments because of the following characteristics:—(1) It produces, with appropriate magnifications, the actual profile of a considerable length of surface. This gives a profile record in which finely spaced irregularities, such as roughness, are superimposed on the more widely spaced irregularities, such as waves or bows. (2) It permits the study of individual irregularities as to size, shape, and relative position with respect to other irregularities.

The method of recording employed consists of plotting the ordinates (heights) of the irregularities as a function of the abscissae. The three basic requirements for successful recording are vertical magnification, horizontal magnification, and a tracing mechanism. A description is given of how these basic requirements have been incorporated in three units, i.e. the tracer, the piloting fixture, and the Amplicorder.

The instrument can be used for inspection of production set-ups for waves and other surface characteristics. As an instrument for use in process engineering, it provides quantitative data for determining the effectiveness of steps taken to improve or obtain certain surface finishes in machining or grinding operations. As a laboratory instrument, it supplies information to those interested in determining the causes, effects, and characteristics of surface irregularities.

SMOKE ABATEMENT

The Illinois Smokeless Furnace.

By J. R. Fellows, A. P. Kratz, and S. Konzo. (From *University of Illinois Bulletin*, Vol. 45, No. 9, September 24, 1947, 80 pages, 28 illustrations.)

The studies that led to the development of the Illinois Smokeless Furnace were begun in 1935, and consisted of experiments with a down-draught conversion burner installed in a hand-fired furnace. As a result of these initial studies the down-draught conversion burner was perfected. They demonstrated the practicability of the principles involved and indicated the possibility of their incorporation in a complete furnace unit. An experimental furnace, designed specifically to facilitate the study of all the factors affecting the performance of a smokeless furnace, was constructed in the Mechanical Engineering Laboratory in the summer of 1939.

The first commercial model was completed in December, 1942. It consists of an integral assembly of a number of parts, including a coking chamber, a coke-burning chamber, a baffle wall, three separate air orifices, and two forms of grates. In the coking chamber each charge of fresh coal is converted to coke, while that converted from the previous charge is burned in the coke-burning chamber at the back. The volatile matter released from the fresh coal mixes with the secondary air introduced through vertical air passages in the baffle wall. The mixture then passes over live coals in the coke-burning chamber, where it is ignited. The burning is completed in the combustion flues and the auxiliary combustion chamber located above the flues. Provision is made in the furnace for the control of the rate of gas release and the air supply for the thorough mixture of gas with secondary air, and for the ignition of the gas-air mixture. In general, the present designs are well adapted to normal production practices involving the fabrication of sheet steel. A study has not yet been made of the modifications necessary to adapt the smokeless principle to cast-iron furnaces or to boilers.

Performance tests of the furnace were conducted in the laboratory to determine the burning characteristics of a wide variety of solid fuels. In general, over a wide range of burning rates, the density of the smoke resulting from the burning of high-volatile coal was comparable with that obtained in using an underfeed stoker, and was well within the limits imposed by existing smoke ordinances.

Data are presented on the desirable proportions of the three sets of air orifices and on the operation of dampers placed over the orifices. The use of a combined check damper and draught regulator together with dampers used in connection with the orifices is recommended. The combined check damper and draught regulator limited the draught at the smoke collar of the furnace to 0·06 in. of water, which in turn limited the temperature that could be attained by the flue gases. Thus the arrangement served as a safety limit control, and it is probable that it could be equally well adapted to a conventional hand-fired furnace.

The performance of the furnace was entirely satisfactory in the Warm-Air Heating Research Residence, from the standpoint both of smokeless operation and of adaptability to temperature control.

THERMODYNAMICS

Dissipation of Heat by Free Convection.

By W. Elenbaas. (From *De Ingenieur, Holland*, Vol. 60, No. 7, February 13, 1948, pp. O.21-O.34, 17 illustrations.)

A body differing in temperature from the surroundings loses energy not only by radiation but also by conduction and convection. The general laws governing this latter dissipation of heat are investigated. The many quantities governing this transfer may be combined into a few dimensionless numbers by means of similarity considerations. This leads to an important simplification and facilitates the survey of experimental and theoretical results. The above-mentioned considerations are first applied to bodies which may be characterised by one linear dimension only (the vertical plate, the horizontal cylinder, the sphere), and next to bodies characterised by two such quantities (the vertical cylinder of finite height, cooling ribs, the inner surface of vertical tubes).

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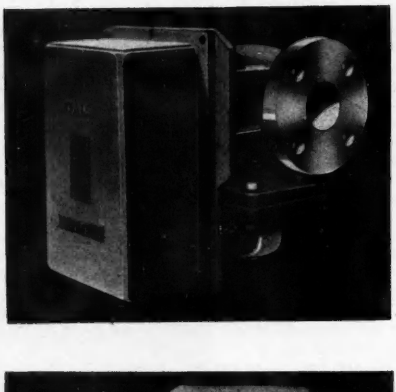

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NEWS OF THE MONTH

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.
Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- PERSONAL
- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- NOTES

PERSONAL.

Mr. F. W. Abraham, M.I.Loco.E., A.M.Inst.T., has been appointed motive power superintendent of the London Midland Region of the Railway Executive.

Mr. E. H. Ball has been appointed a director of the Metropolitan-Vickers Electrical Co., Ltd., Trafford Park, Manchester.

Dr. H. Roxbee Cox has been elected President of the Royal Aeronautical Society; **Sir John Buchanan, Mr. W. G. A. Perring** and **Mr. N. E. Rowe** have been elected Vice-Presidents.

Mr. I. C. Forsyth, M.I.Loco.E., has been appointed works manager of the Crewe Locomotive Works.

Mr. J. W. Garton has been appointed chairman of Brown Bayley's Steel Works Ltd., Sheffield 9.

Mr. Charles Greenwood, M.I.C.E., City Engineer and Surveyor of Chester, has been elected President of the Institution of Municipal Engineers, which celebrates this year the 75th anniversary of its formation.

Mr. T. Lawrie has been appointed general manager, **Mr. W. D. D. Fenton,** secretary, and **Mr. H. W. Simpson,** deputy secretary of the North of Scotland Hydro-Electric Board. 16 Rothesay Terrace, Edinburgh 3.

Mr. H. A. Longden, B.Sc. (B'ham), A.M.I.C.E., F.G.S., general manager, No. 8 (Castleford) area, North-Eastern Division (Yorkshire) National Coal Board, has been appointed production director of the Division.

Mr. W. Lyons, chairman and managing director of Jaguar Cars Ltd., Coventry, has been elected vice-president of the Society of Motor Manufacturers and Traders Ltd.

Mr. W. J. Price has been appointed general sales manager of Birmingham Aluminium Casting (1903) Co., Ltd., Smethwick 40, Staffs., in succession of **Mr. E. W. Sanger** who joined the Begwaco Meters Ltd., Farnworth, Lancs., as joint managing director

Dr. S. C. Redshaw, M.Sc., M.I.C.E., F.R.Ae.S., chief engineer, and **Mr. G. C. Haynes,** secretary of Boulton Paul Aircraft Ltd., have been elected directors of the Company.

Mr. J. W. Rutty has been appointed manager for Australia of the W. T. Henley's Telegraph Works Co., Ltd., Australian head office which has been transferred from Sydney to Melbourne.

Mr. J. R. Rylands, M.Sc., M.I.Mech.E., M.I.E.E., chief engineer of E. Green & Son Ltd., Economiser Works, Wakefield, has been elected a director of the Company.

Dr. I. G. Slater has taken up the position of Director of Operational Research at the Admiralty, London.

Dr. Norman Swindells, M.A., has been appointed chief metallurgist to McKechnie Brothers, Ltd., Rotton Park Street, Birmingham 16.

Mr. R. H. Stephenson, O.B.E., director of Smith's Dock Co., Ltd., North Shields, Glasgow, has been re-elected chairman, and **Mr. G. S. Cromar, C.B.E.,** of Barclay, Curle & Co. Ltd., and **Mr. E. L. Champness,** of Palmers Hebburn Co., Ltd., Hebburn-on-Tyne, have been elected vice-chairmen of the Dry Dock Owners and Repairers Central Council.

Mr. R. J. Turner has been appointed general manager, industrial sales, **Mr. J. A. V. Watson, O.B.E.,** general manager, Castrol sales, and **Mr. A. A. Barr,** publicity manager of C. C. Wakefield & Co., Ltd., 46 Grosvenor Street, London, W.1.

Mr. F. Wardrobe, A.M.I.E.E., has retired from the position of chief engineer of William Jessop & Sons Ltd., Sheffield 1, but remains a director of the company and consultant on plant and engineering developments.

Mr. F. Woodfield, director and works manager of The Park Gate Iron & Steel Co., Ltd., Rotherham, and President of the Iron & Steel Trades' Employers' Association, sailed for San Francisco via New York, to attend the International Labour Conference as one of the advisers appointed by the British Employers' Confederation.

Mr. Eric G. Yarrow, M.B.E., and **Mr. G. Sturrock Brown, B.Sc.,** have been elected directors of Yarrow & Co., Ltd., Scotstoun, Glasgow.

OBITUARY.

The death occurred at his home in Bristol, of **Mr Alfred W. Bennis, M.I.Mech.E.,** the chairman and managing director of Bennis Combustion Ltd., of Little Hulton, near Bolton.

Alexander Thomas Gillanders, director of Southern United Telephone Cables Ltd., and of Union Cable Co., Ltd., former power cable manager of Standard Telephones & Cables Ltd., died suddenly at his home in Eltham at the age of 65.

Mr. E. A. J. Koch, well-known industrial consultant, of Ashtead, Surrey, died recently at the age of sixty.

Sir W. Murray Morrison, a pioneer of the aluminium industry in Great Britain, died in London at the age of 74.

Mr. Sydney Smith who had been connected with the Wellman companies for many years, and little more than twelve months ago succeeded his brother, the late Mr. James Foster-Smith in the chief executive position as managing director of The Wellman Smith Owen Engineering Corporation, Ltd., Parnell House, Wilton Road, London, S.W.1, died on 30th May, 1948, after a period of failing health.

BUSINESS NOTES.

Rawsthornes (Bolton) Ltd. will shortly be removing to larger premises. The change has been made to consolidate both the Boiler and Pipe Covering, and the Felt Washer and Packings Departments of the business. The new address will be Rawsthornes (Bolton) Limited, Bullock Street Works, Bolton, Lancs.

Hordern, Mason & Edwards, Ltd., Birmingham 24, have engaged, due to continual expansion of business, two additional technical representatives. **Mr. C. H. Bennett,** well known in the power press industry as an expert on tools and production, will operate from the Works to devote his attention to the Midlands, and **Mr. A. B. Paget** who has wide experience of press works with principal users in both Britain and in the U.S.A., will attend to the North of England from his temporary address at 130 Sandy

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Lane, Lower Darwen, near Blackburn, Lancs. Tel.: Blackburn 49 and 145.

The British Aluminium Co.'s North Western Area Branch Office has been removed to 46 Fountain Street, Manchester 2. Tel.: Deansgate 3639.

Dunlop Special Products Ltd., have removed their headquarters to 19-20 New Bond Street, London W.1. Tel Mayfair 9040.

Export of Roller Shutters. Mr. R. R. Seymour, managing director of G. Brady & Co., Ltd., Manchester 4, has left for Canada and the United States where he will make an extensive business tour in a specially designed car fitted with roller shutters. The journey is being undertaken for the purpose to earn dollars by exporting roller shutters and the new shutter doors made from anodized non-ferrous metal in a variety of color schemes.

The Engineering and Marine Exhibition, 1949. The whole of Olympia, London, has been secured for the next Exhibition which will be held from August 25th to September 10th, 1949. The addition of the Empire Hall will make possible the acceptance of a certain number of new exhibitors.

Royal Academy Exhibition of Industrial Design. The first exhibition of the work of Royal Designers for Industry will be held at Burlington House, London, next October. The work exhibited will include the famous De Havilland jet aircraft designed by Sir Geoffrey De Havilland, C.B.E.; the Coronation Scot and the Vickers Viking airliner by Brian O'Rorke; and equipment for London Transport by Charles Holden. The distinction "Royal Designer for Industry" (R.D.I.) may at no time be held by more than 40 designers. A limited number of foreign designers hold the title of Hon. R.D.I. They are: Alvar Aalto of Finland; Walter A. G. Gropius, M. McKnight Kauffer, and Raymond Loewy of the U.S.A.; Edward Hald of Sweden; and Steen Ejler Rasmussen of Denmark.

Air Commodore Whittle is Awarded £100,000. The Treasury and the Ministry of Supply have accepted the recommendation of the Royal Commission on Awards to Inventors that a total award of £100,000, free of tax, should be made to Air Commodore Frank Whittle, C.B., C.B.E., in respect of his achievements in the science of aeronautics by devising and developing practical means of applying the principle of jet propulsion and for his improvements in the designs of gas turbines and air compressors for use in connection with jet propulsion.

A Research Scholarship of £400 per annum and tenable for two years has been offered by the **Aluminium Development Association** to encourage and facilitate research in the application of light alloys to ship construction. It will be administered by a Committee of the Institution of Naval Architects and it is hoped to make the award in September, 1948. Full particulars of entry, which closes on July 31st, can be obtained from the Secretary, Institution of Naval Architects, 10 Upper Belgrave Street, London, S.W.1.

National Foundry College. The next session will open on 27th September, 1948, and continue until the end of July, 1949. Those interested in the Course can obtain a form of application for admission, together with the Prospectus and Curriculum from the Head of the College, Mr. J. Bamford, B.Sc., at the Technical College, Wultruna Street, Wolverhampton.

"The Nickel Bulletin." The March issue of "The Nickel Bulletin" contains an article by R. H. Craig, B.Sc., A.M.I.Mech.E., A.M.I.E.E., on the mechanical engineering aspects of the development and manufacture of the Cathode Ray Tube. Among the abstracts published, those dealing with electrodeposition and other coating methods will be of considerable interest in view of the current trend in all countries towards the use of thicker electrodeposited coatings. The April edition draws further attention to the subject with an article on the value and specifications for nickel electroplate.

The abstracts on heat-resisting alloys will be of particular interest in view of the new developments in gas turbine automobile engines.

"The Nickel Bulletin" is obtainable free of charge from The Mond Nickel Company Limited, Grosvenor House, Park Lane, London, W.1.

Calibration of Carburettor Jets. The second revision of B.S. 720, Calibration of Carburettor Jets has just been published.

Originally prepared to cover carburettor jets for aircraft and automobile engine, the standard now applies to jets for all types of engines and the maximum fuel flow that can be dealt with under this standard has been raised from 2,000 to 15,000 ml. per minute. Price 3s.

Jet-Prop Flight-Tested on Air Routes. The development of British turbo-prop engines—a gas-turbine jet engine driving a propeller—has advanced another stage by the introduction of the *Theseus-Lincoln* into R.A.F. Transport Command, where it has embarked on a series of extensive flight trials on the trunk routes between England and the Middle East, carrying freight.

The Bristol *Theseus* is therefore the first prop-jet engine to operate on an established air route. It was also the first engine of its kind to pass the strict 150-hour Type Test.

To speed up flight-testing of the engine, a standard Avro *Lincoln* four-engined bomber was handed over to the Bristol company for the installation of two *Theseus* engines in place of the two outer *Merlins*. In this form, the *Theseus-Lincoln* was flown for the first time on 17th February, 1947, by the company's chief test pilot, Mr. "Bill" Pegg. It was demonstrated at the Society of British Aircraft Constructors' Display last year and will be flown at the Society's Display in September, when the public will be able to see this and other prop-jet types.

A second *Theseus-Lincoln* has been completed and is joining the first in Transport Command.

Meanwhile, the Handley Page *Hermes V* airliner, which will be powered by *Theseus* engines, is now under construction and is expected to fly early next year. Production aircraft will follow and will take their place, probably by 1951, as the first prop-jet airliners in the world to be put into regular service.

Thus the British aircraft industry, working with the Ministry of Supply and the Air Ministry, will have taken considerably less than ten years to have originated, developed and put into commercial service a completely revolutionary type of aero-engine.

The Aluminium Wire & Cable Co., Ltd., 10, Buckingham Place, London, S.W.1, announce that they have now transferred to their works at Swansea the manufacturing facilities for aluminium and alloy wire and hot rolled rod previously operated by The British Aluminium Company Ltd., and Reynolds Light Alloys Ltd.

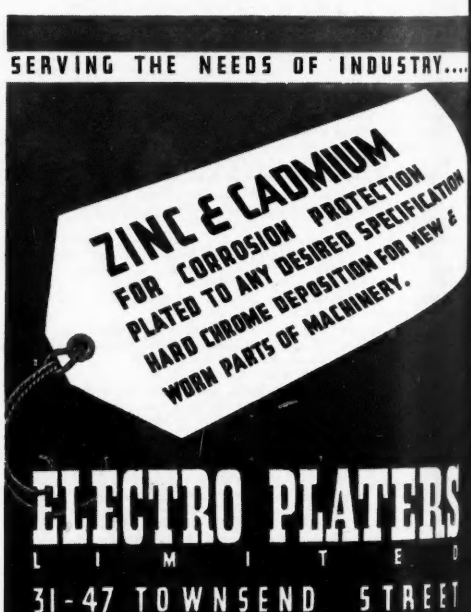
Sales of wire and rod made at Swansea will continue for the present to be handled by The British Aluminium Company Ltd., but will be taken over by the Aluminium Wire & Cable Co., Ltd., in a few months time. The exact date will be announced later but in the meantime, deliveries to customers against existing orders will not be affected.

BOOKS.

Electric-Motor Control Gear. By J. L. Watts. Publisher: Electrical Review, Dorset House, London. S.E.1. 84 pp. Price 5s.

"Electric-motor Control Gear" explains from a practical viewpoint the principles underlying the construction and operation of the various types of apparatus available for the purpose. Since most electrical failures associated with motor drives originate in the control gear, which also affects the starting torque and safe use of the machines, the book should be of very considerable help to engineers responsible for installation and maintenance and to all concerned with the efficient performance of electric motors.

"Forming of Austenitic Chromium-Nickel Stainless Steels." With the progress in the development of industrial equipment, more and more problems are encountered in the forming and fabricating of parts of intricate design. The use of stainless steels in many industries is growing rapidly. A new book, "Forming of Austenitic Chromium-Nickel Stainless Steels," was compiled to give fabricators of metal equipment a better under-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standing of the exceptional adaptability of stainless steels to all modern processes of forming. This book presents a detailed description of the modern forming procedures as applied to chromium-nickel stainless steels and as practiced in the fabrication plants of the United States. Bending and straight flanging; forming of curved sections and tubing; deep drawing; die forming; forming of contoured-flanged parts, and forming by miscellaneous methods are some of the methods discussed in the book.

The specific examples of forming technique are supplemented by details of tool design and tool materials, lubricants, data on dimensions, and consecutive steps in fabrication.

The unexpected demand for the book has exhausted the initial printing and copies of the second printing, as long as they are available, may be reserved at $4.00 through The International Nickel Company, Inc., 67 Wall Street, New York 5, N.Y., or The International Nickel Company of Canada, Limited, 25 King Street West, Toronto 1, Ontario.

Proceedings of the Society for Experimental Stress Analysis, Vol. V., No. 2, just published by Addison-Wesley Press Inc., Kendall Square, Cambridge 42, Mass., U.S.A. Price $6, contains the following articles:

The Measurement and Interpretation of Post-Yield Strains; On the Removal of Time Stresses in Three-Dimensional Photoelasticity; The Development of a Technique for Testing Stiff Panels in Edgewise Compression; An Evaluation of Some Approximate Methods of Computing Landing Stresses in Aircraft; Equivalence of Photoelastic Scattering Patterns and Membrane Contours for Torsion; A Three-Component Force Recorder; Three-Dimensional Photoelastic Analysis by Scattered Light; Stress Distribution around a Hole near the Edge of a Plate under Torsion; Response of Damped Elastic Systems to Transient Disturbances; Electric Strain Gage Analysis of a Fifty-Foot Hortonsphere; A Pendulum Analyzer for Mechanical Transients; A Strain-Gage Balance System for a Supersonic Wind Tunnel; The Photo-Grid Process for Measuring Strain Caused by Underwater Explosions; Some Evaluations of Stresses in Aneroid Capsules; Allowable Stresses for Steel Members of Finite Life; Theoretical and Experimental Investigation of Buckling Shock Mount.

Quality Control in Production. By H. RISSIK. With a Foreword by Sir Frank Gill. Publisher: Sir Isaac Pitman & Sons Ltd., Parker Street, London, W.C.2. 180 pp. with 58 figures, tables and charts. Price 21s.

Quality control—the statistical method of controlling product quality during manufacture—has proved its value by eliminating faulty products, reducing process inspection, and facilitating the flow of production. These advantages are especially important to-day because of the urgent need for large output without waste of materials, man-hours and machines. Rissik's book is a complete manual on the subject.

The Electricity Supply Acts. By JAMES W. CHANT. Publisher: Stevens & Sons Ltd., 119-120, Chancery Lane, London, W.C.2. 262 pp. Price 25s.

The Electricity Act, 1947, considerably amends the Electricity Supply Acts, including those relating to the North of Scotland. All these Acts as amended, together with other relevant Statutes, are presented in one handy volume.

Automobile and Aircraft Engines. Volume I. The Mechanics of Petrol and Diesel Engines. Fourth Edition. By Arthur W. Judge. Publisher: Sir Isaac Pitman & Sons Ltd., Parker Street, London, W.C.2. 307 pp. with 223 figures and 18 tables. Price 25s.

Since 1936 when the third edition of Judge's important book on I.C. engines was published important progress has been achieved in various aspects of petrol and diesel engines. The present, fourth edition, has accordingly been revised and considerably extended, much of the earlier material being either deleted or rewritten. Additional sections on engine vibration, engine mountings, torsional vibration, balancing and rotating members, modern balancing machines, and valve cams and followers have been included. R. G. Manley contributed a chapter on Torsional Oscillations in Engines.

Chemical and Electro-Plated Finishes. By H. SILMAN. With a Foreword by H. Moore. Publisher: Chapman & Hall Ltd., 37, Essex Street, London, W.C.2. 414 pp. with 131 figures and 12 tables. Price 30s.

Silman's excellent book gives a rather comprehensive account of modern industrial finishing processes, together with the chemical and physical principles involved in them. The value of the book is enhanced by the incorporation of particulars of the latest American developments.

Fundamentals of Engineering Drawing. By WARREN J. LUZADDER. Publisher: George Allen & Unwin Ltd., 40, Museum Street, London, W.C.1. With over 1000 figures. Price 30s.

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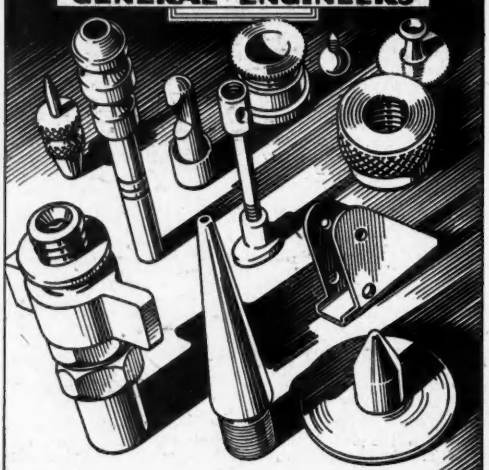

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